USAAVLABS TECHNICAL REPORT 67-4

INVESTIGATION OF SOLID LUBRICANTS FOR HELICOPTER TRANSMISSIONS

By

Paul H. Bowen

March 1967

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-307(T)
WESTINGHOUSE ELECTRIC CORPORATION
RESEARCH LABORATORIES
PITTSBURGH, PENNSYLVANIA

U S ARMY MATERIEL COMMAND

Distribution of this document is unlimited

81

Disclaimers

The findings in this report are not to be construed as an official Department of the Army position unless so designated by other authorized documents.

When Government drawings, specifications, or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the Government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation, or conveying any rights or permission, to manufacture, use, or sell any patented invention that may in any way be related thereto.

Trade names cited in this report do not constitute an official endorsement or approval of the use of such commecial hardware or software.

Disposition Instructions

Destroy this report when no longer needed. Do not return it to originator.

DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 23604

The objective of this program was to determine the feasibility of using a dry lubricant in present helicopter transmissions as an auxiliary means to extend the catastrophic failure time to a minimum of 30 minutes.

This report presents the results of a literature survey, preliminary screening tests of solid lubricants, and laboratory testing of bearings and gears under speeds and loads experienced in helicopter transmissions, simulating both normal and failed lubricating conditions.

This command concurs in the conclusions made by the contractor.

Task 1M125901A01410
Contract DA 44-177-AMC-307(T)
USAAVLABS TECHNICAL REPORT 67-4
March 1967

INVESTIGATION OF SOLID LUBRICANTS FOR HELICOPTER TRANSMISSIONS

By

Paul H. Bowen

Prepared By

Westinghouse Electric Corporation
Research Laboratories
Pittsburgh, Pennsylvania

For

U.S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

SUMMARY

The feasibility of using solid lubricants in bearings and gears has been demonstrated as a satisfactory method for preventing catastrophic failure and for providing emergency operation of helicopter transmissions in the event of an oil lubrication failure.

Modified size-206 Conrad-type ball bearings with retainers of a glass reinforced polyimide, WRP-140, operated initially under a base line condition of oil lubrication for 40 hr. This was followed immediately by 0.5 hr of residual oil lubrication and 1.5 hr with no external lubrication at 14,000 rpm for various thrust loads of up to 800 lb without failure. Bearings with a silver alloy - Teflon composite, RB-HP-15 operated under similar oil and residual lubrication for 0.5 hr at loads of up to 450 lb before failure.

A conventional 12 diametral pitch (DP) AISI 9310 gear set, using a 6-in. gear and a 2-in. pinion with a 2.5-in. idler of WRP-140, operated under base line conditions of oil lubrication for 40 hr at approximately 14,000 rpm and at a tooth load of 160 lb [640 lb/in. tooth width (ppi)]. Similar gears with the same idler operated 0.5 hr with no external lubrication at a load of 1220 ppi (305 lb tooth load). With an RB-HP-15 idler, other gears were operated without lubrication for 0.5 hr at a load of 1220 ppi.

A literature survey was made of solid and fluid lubricants; helicopter transmissions and related components, such as gears and bearings; solid lubricant specifications; and other consolidated references on solid lubricants.

PREFACE

This program was conducted under U.S. Army Aviation Materiel Laboratories Contract DA 44-177-AMC-307(T), with Mr. E. Rouzee Givens as Project Engineer.

The program at the Westinghouse Research Laboratories was directed by Mr. P. H. Bowen. Mr. E. S. Bober, Manager, Lubricants and Electrochemical Technology, R&D, provided management guidance. Dr. J. H. Freeman and Messrs. E. J. Traynor, D. J. Boes, K. W. Grossett, J. R. McDowell, and J. Valentich contributed valuable consultation on organic composites, solid lubricants, gears, and the torque telemetering system used in the feasibility program. Technical assistance was rendered by Messrs. H. R. Wilkinson and G. R. Kelecava.

BLANK PAGE

TABLE OF CONTENTS

TABLE OF CONTENTS (CONT'D)

APPENDIXES

ILLUSTRATIONS

TABLES

INTRODUCTION

Adequate lubrication of the gears and bearings in the power transmission of rotary-wing aircraft is critical to its proper operation. There are few comparisons in other aircraft where reliability is so vital a factor. The occurrence of a lubrication system failure of the transmission drive system would cause catastrophic failure of the related gears or bearings and result in stoppage of the rotor. In a combat situation, this failure would result in an immediate abort of the mission, with loss of the aircraft and probable loss of personnel.

Research efforts have shown that solid lubricants may be considered as candidate materials in providing adequate lubrication to both gears and bearings without the necessity of fluid lubricants. Johnson[1] determined wear and friction of composites containing molybdenum disulfide in 1957. Since that time other investigators made significant progress in the development and application of solid lubricants. Some of this work is mentioned in the "Discussion".

The program reported herein was conducted to determine the feasibility of using solid lubricants in gears and bearings under speeds and loads experienced in helicopter transmissions to simulate both normal and failed lubricating conditions. Lubricating oil MIL-L-7808, a diester sebacate fluid, was used as the oil for 40 hr of conventional operation; the solid materials provided lubrication in the emergency condition for an additional 0.5 hr, when no oil was supplied. Extending operation of the bearings and gears for 0.5 hr after loss of oil, through battle damage of the transmission, should be sufficient to allow the helicopter to return to friendly territory or to select the location for an emergency landing.

An experimental evaluation program was conducted to screen selected solid lubricating materials, to evaluate these materials as lubricants for both bearings and gears, and to perform compatibility tests. An additional part of the evaluation program covered the method used in determining the bending loads and Hertz stresses of the bearings and gears. This determination is covered in Appendix I.

A literature survey covered a review of abstracts of scientific literature and research programs concerning solid lubricating materials, and systems, fluid lubricants and systems, mechanisms, gears, bearings, and solid lubricating materials with related specifications. Some of the data are presented in Appendixes II and III.

EXPERIMENTAL RESULTS

SCREENING TESTS

Fifteen solid lubricating materials were selected as possible candidates that could be used with gears and bearings to provide emergency operation in the event of an interrupted oil supply. Three conditions of lubrication were evaluated and covered:

1. Circulating Oil Operation
2. Residual or Marginal Operation
3. Dry Operation

A summary of the lubricants selected and a description of the lubricating conditions are noted in Table I.

TABLE I

SOLID LUBRICANT MATERIALS

Material Designation	Composition (wt %)
WRP-140	60% polyimide; 40% glass
RB-HP-15	85% Ag-Hg; 15% PTFE
101	53% WSe_2; 12% Co; 35% Ag
108	80% MoS_2; 20% Ta
114	50% MoS_2; 38% Ta; 12% Fe
RB-AP-1	70% Ag; 20% PTFE; 10% WSe_2
RB-CP-1	60% Cu; 30% PTFE; 10% WSe_2
RB-GA-IN	80% WSe_2; 20% GaIn
Thin Film	Na_2SiO_3; MoS_2
2136(AF-I-L-MIO)	90% Ni; 10% MoS_2
2137(AF-IL-W20)	80% Ni; 20% WS_2
2138(AF-IL-65)	95% Ni; 5% CaF_2
RBP	40% Brz; 40% polymide; 20% graph
WRC-92M	70% asbestos; 30% BPO_4
WRW-1	90% graphite (cloth); 10% polyimide

The solid lubricants were screened in a wear and friction tester under various loads and sliding conditions. The loads ranged from 3000 psi under fluid conditions to 200 psi for dry operation. The sliding velocity of 2500 ft/min represented the sliding conditions experienced in a size-206 ball bearing and on the tooth face of a 2-in. gear rotating at 14,000 rpm.

Circulating Oil Operation

The solid lubricating material specimens in the form of 0.25 in. x 0.75 in. x 0.50 in. blocks were screened at loads of 1500 psi and 3000 psi in the wear and friction tester. The 0.25 in. x 0.75 in. face of the specimen rubbed against the periphery of a 0.25-in.-wide rotating Type 440-C stainless steel disk.

The contact surfaces of the specimen and disk were flooded with oil to act as a lubricant and coolant. The oil feed of 10 drop/min on the disk used in the tests was sufficient to provide hydrodynamic lubrication under most circumstances.

Three of the candidate materials failed during preliminary tests and were not included in any of the further test programs. The WRW-1 impregnated graphite cloth and the WRC-92M phosphate composite had poor strength properties. The RBP bronze-graphite filled plastic material exhibited poor wear characteristics under dry operating conditions.

The average results of multiple tests of the remaining twelve materials are shown in Table II. All of the materials, with the exception of the

TABLE II

CIRCULATING OIL SPECIMEN SCREENING AT HIGH LOADS

Load: 1500 psi
Sliding velocity: 2500 ft/min
Time: 0.5 hr

Material	Coef. of Friction	Wear (MM)	Stabilized Temp. (°F)	Remarks
WRP-140	0.09	Polish	121	Stable operation
RB-HP-15	0.07	0.5	115	Stable operation
101	0.08	0.4	86	Stable operation
108	0.02	0.5	106	Stable operation
114	0.01	1.0	87	Marginal wear
RB-AP-1	0.05	0.7	125	Stable operation
RB-CP-1	0.06	0.5	120	Stable operation
RB-GA-IN	0.05	0.3	84	Stable operation
Thin Film	0.09	1.1	93	Film worn through
2136	0.08	1.1	210	Variation in wear
2137	0.06	0.4	80	Stable operation
2138	0.09	1.2	200	Variation in wear

thin film specimen, exhibited satisfactory to marginal lubricating characteristics at a load of 1500 psi. The coefficient of friction measured was extremely low and indicated 11 of these materials could be used as

sleeve bearings under partial hydrodynamic lubricating conditions. Wear was uniformly low for all of the materials. The operating temperatures varied over a wide range for the different materials. Part of the temperature variation was attributed to the differences in thermal conductivity of the materials and part to the location of the thermocouple in the test specimen. The molybdenum disulfide resin bonded thin film material exhibited low wear, friction, and operating temperature even though the film had worn through and steel was rubbing against steel. It appears that only a small amount of solid lubricant in addition to the oil is required to provide excellent lubrication under the fluid test conditions, even at rather high loads.

Repeat tests were made on the same group of twelve materials at a load of 3000 psi. The results of these multiple tests are shown in Table III.

TABLE III

CIRCULATING OIL SCREENING AT VERY HIGH LOADS

Load: 3000 psi
Sliding velocity: 2500 ft/min
Time: 0.5 hr

Material	Coef. of Friction	Wear (MM)	Stabilized Temp. (°F)	Remarks
WRP-140	0.10	1.5	232	Variation in temperature
RB-HP-15	0.06	0.6	175	Stable operation
101	0.23	0.8	121	Variation in friction
108	0.05	1.1	133	Stable operation
114	----	---	---	Test not completed
RB-AP-1	0.06	0.6	150	Stable operation
RB-CP-1	0.06	0.6	144	Stable operation
RB-GA-IN	0.09	0.4	109	Stable operation
Thin Film	----	---	---	Test not completed
2136	0.26	2.1	220	Marginal operation
2137	0.29	0.6	175	Variation in friction
2138	0.27	2.2	230	Marginal operation

The materials exhibited a greater variation in both friction and wear when compared to the results obtained at 1500 psi. The WRP-140 material exhibited a variation in temperature from test to test. The temperatures ranged from 120 to 232°F while the wear and friction values remained relatively constant. A somewhat similar variation occurred in the friction for the 101 and 2137 materials. Three of the materials (2136, 2137, and 2138) exhibited rather high friction characteristics. This may be the result of the change in specimen surface as it wears in the transfer of specimen material to the disk. For either condition, the effect was the same: hydrodynamic lubrication was lost, and boundary layer lubrication became predominant.

Residual, or Marginal Operation

The residual oil tests were run on the same tester to evaluate the materials under initial emergency conditions.

Oil was supplied to the surfaces of the solid lubricant and metal, and then the flow was stopped. Only that oil that remained in the system was available for lubrication. The oil supply was exhausted within several minutes of operation. Repeat tests were made using specimens that had been immersed in MIL-L-7808 oil for 45 days to determine if the oil would adversely affect the lubricating properties of the solid materials.

The results of this test group are shown in Table IV. In general, the materials exhibited higher friction, wear, and operating temperatures than

TABLE IV

RESIDUAL OIL SPECIMEN SCREENING

Load: 150-200 psi
Sliding Velocity: 2500 ft/min
Time: 0.5 hr
Condition: After 45-day oil immersion

Material	Coef. of Friction	Wear (MM)	Stabilized Temp. (°F)	Remarks
WRP-140*	0.09	0.2	109	Stable operation
RB-HP-15*	0.26	1.9	198	Stable operation
101*	0.08	6.3	178	Marginal wear
108*	0.08	2.6	130	Stable operation
114*	0.18	3.4	130	Stable operation
RB-AP-1	0.22	2.0	176	Stable operation
RB-CP-1	0.26	2.0	175	Stable operation
RB-GA-1N	0.16	3.1	135	Stable operation
Thin Film	----	---	---	Not tested
2136	0.41	12.0	420	Marginal operation
2137	0.47	8.0	290	Marginal operation
2138	0.46	6.0	235	Marginal operation

*Indicated materials were screened again after 90 days with similar results

that of the circulating oil tests even though the loads were reduced to 150-200 psi. The most desirable materials were: WRP-140, 108, RB-AP-1, and RB-HP-15.

The WRP-140 material exhibited low friction and wear because of the absorbed oil. The material acted as a wick and provided sufficient oil at the rubbing interface to provide satisfactory lubrication. The 108 material provided satisfactory lubrication because of the extremely high MoS_2 powder content. The material had sufficient strength and hardness properties to provide relatively low wear rates under test conditions. The RB-HP-15 material, which contained high percentag of silver, contained Teflon, which aided in the transfer of a silver film to the steel disk. However, this transfer apparently required more energy than the 108 material and resulted in a correspondingly higher friction and operating temperature.

Dry Operation

The screening wear and friction tests were made on twelve specimens, without benefit of any fluid lubrication, under the same load and speed conditions as the residual oil lubrication tests. The average results of the multiple tests are shown in Table V. The five materials known as WRP-140, RB-HP-15, 108, 114 and RB-CP-1 appeared to be satisfactory; they exhibited somewhat similar wear and friction characteristics.

TABLE V

DRY SPECIMEN SCREENING

Load: 150-200 psi
Sliding velocity: 2500 ft/min
Time: 0.5 hr

Material	Coef. of Friction	Wear (MM)	Stabilized Temp. (°F)	Remarks
WRP-140	0.28	4.7	186	Tested at loads up to 1500 psi
RB-HP-15	0.26	4.8	170	Very stable operation
101	0.13	7.8	220	Excessive wear
108	0.14	2.8	150	Stable operation
114	0.13	2.8	146	Stable operation
RB-AP-1	0.27	4.7	160	Max. temp., 320°F
RB-CP-1	0.25	2.8	136	Stable operation
RB-GA-IN	0.34	4.6	232	Failed after 5 min.
Thin Film*	0.09	4.8	260	Failed after 5 min.
2136*	0.60	8.0	330	Failed after 2 min.
2137*	0.76	2.0	350	Failed after 1 min.
2138*	0.73	8.0	290	Failed after 1 min.

*Wear and temperature at time of failure (average of 20 tests)

It was observed during the screening tests that some of these five materials lacked other properties that were necessary for satisfactory bearing

lubrication. The 108 and 114 materials were similar to carbon materials and were extremely brittle with little shock resistance. The RB-CP-1 material exhibited low strength properties and was deleted from the subsequent test program. Two solid lubricant composites, WRP-140 and RB-HP-15, were selected as candidate materials for the functional bearing and gear tests.

BEARING TESTS

The test bearings were 206-size, 30mm, deep-groove Conrad-type using a solid lubricated retainer. They were evaluated at 14,000 rpm in a special bearing tester. The test apparatus used either one or two test bearings on a horizontal spindle. The load was applied in an axial direction. A complete description of the test bearings, tester, and test procedure is given in the section entitled "Experimental Program".

The bearings were modified in order to use the solid lubricant retainers. Each retainer was identified by a suffixal number as part of the composite material identification. Three retainers were used in the test program. The dimensions of these retainers: RB-HP-15-1, WRP-140-1 and WRP-140-2, were similar. The WRP-140-1 retainer, incorporated a 0.040-in. thick steel reinforcing plate on each side to provide additional impact strength. One of the retainers is shown in Figure 1.

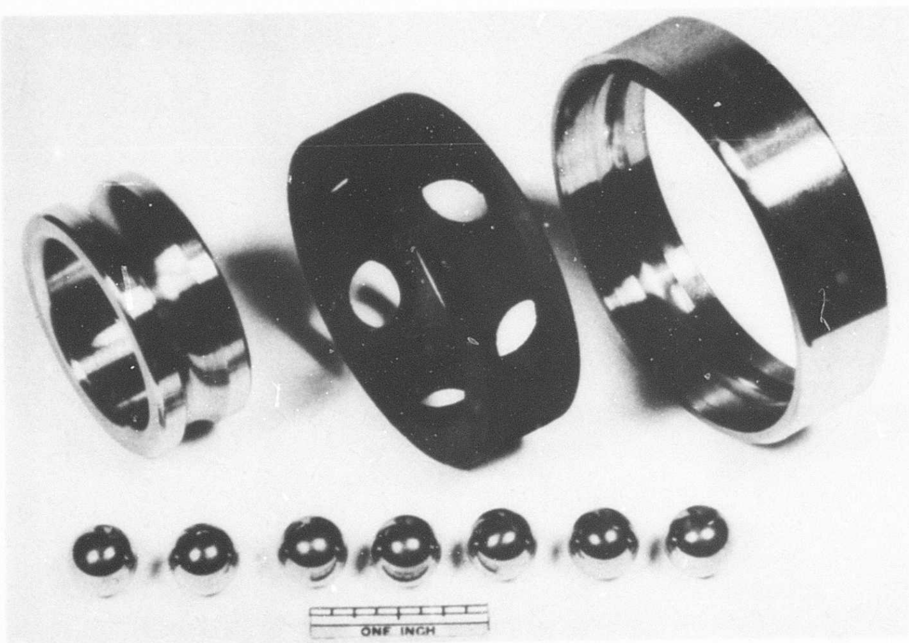


Figure 1. Ball Bearing With WRP-140-2 Retainer

The bearings were evaluated under conditions of (1) circulating oil operation, (2) residual or marginal oil operation, and (3) dry operation.

Circulating Oil Operation

The first series of 40 hr duration tests was made in circulating oil at a speed of 14,000 rpm on bearings with the WRP-140-1, WRP-140-2 and RB-HP-15-1 retainers. Thrust loads of 225 lb and 450 lb were used on each bearing. In addition, comparison tests were run where bearings with pressed steel retainers were used. Each bearing was isolated and used an individual oil supply and drain system. The normal oil flow to each bearing ranged from 450 to 520 cc/min.

The results of these tests are shown in Table VI. The bearings were evaluated in groups of two, with the first listed bearing of each set being located in the front of the tester. All of the bearings operated satisfactorily, and no evidence of component or retainer wear was apparent. The internal clearance of the test bearings was approximately 3 to 6 times that of commercial bearings with pressed steel retainers. This greater clearance was not required for the circulating oil operation but was desired for the dry operation. These results indicate that the increased clearance was not detrimental to circulating oil operation with regard to operating temperature and torque, as shown in Figure 2 and Figure 3.

The weight loss of the WRP-140-1 and -2 retainers is an estimate. The retainers lost or gained weight as a function of the amount of oil used. The amount of retainer wear was insignificant. Every effort was made to remove the oil and to weigh the retainers before and after test under the same condition.

Residual or Marginal Operation

The residual or marginal lubrication tests were made to evaluate operation of a bearing after the oil flow was stopped to simulate a lubrication system failure. In this test series, the bearings with composite retainers were evaluated in the front location. A facility bearing with a pressed steel retainer was located in the rear bearing position.

Separate tests were made for each of the RB-HP-15-1 and WRP-140-2 retainers at both 225- and 450-lb thrust loads at 14,000 rpm. Each test was of 1 hr duration. During the first 0.5 hr that the bearing operated, the circulating oil system was used to obtain stabilized bearing and oil temperatures. The oil flow was stopped to the test bearing, and the test continued for another 0.5 hr. Oil flow was supplied continuously to the rear facility bearing during the entire test. The results of test are shown in Table VII.

The bearings satisfactorily completed the tests, and no visual evidence of wear was noted on either the bearing components or the retainers.

TABLE VI

CIRCULATING OIL BEARING OPERATION

Speed: 13800-14000 rpm
Oil Flow: 450-520 cc/min per brg
Time: 40 hr

Thrust Load (Lb)	Bearing Retainer	Temperature (°F) Brg.	Oil	Static* Torque (In.-Oz.)	Weight Loss (gm) Retainer	Ball	Brg. Clearance (In.) Before	After	Remarks
450	Pressed Steel Pressed Steel	139 142	135 135	24	ND** ND	ND ND	0.0007 0.0007	NC** NC	Both in excellent condition. No visible wear.
225	RB-HP-15-1 Pressed Steel	137 140	134 134	18	0.003 ND	0.002 ND	0.0030 0.0007	NC NC	Both in excellent condition. No visible wear.
225	WRP-140-1 Pressed Steel	130 140	127 127	16	0.001 ND	0.002 ND	0.0042 0.0008	NC NC	Both in excellent condition. No visible wear.
450	RB-HP-15-1 Pressed Steel	144 142	132 132	20	0.003 ND	0.001 ND	0.0036 0.0009	NC NC	Both in excellent condition. No visible wear.
450	WRP-140-2 Pressed Steel	140 142	130 130	17	0.002 ND	0.001 ND	0.0024 0.0008	NC NC	Both in excellent condition. No visible wear.

*Total static torque for both bearings
**Not determined or no change

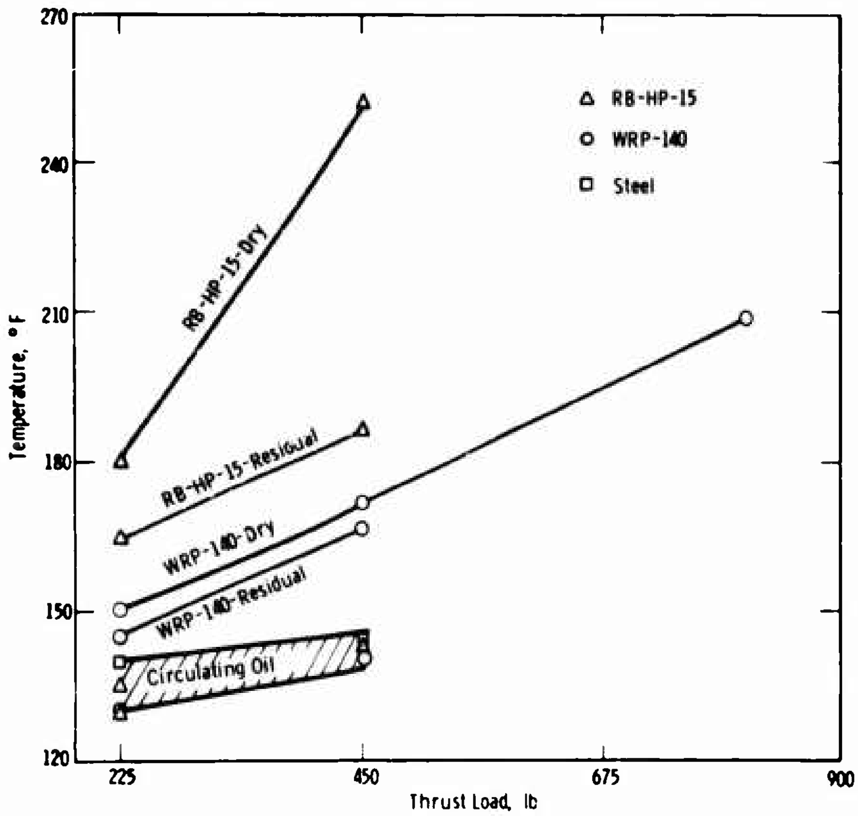
TABLE VII

RESIDUAL OIL BEARING OPERATION

Speed: 13800-14000 rpm
Time: 0.5 hr fluid + 0.5 hr no fluid

Thrust Load (Lb)	Bearing Retainer	Bearing Temp. (°F)	Static* Torque (In.-Oz.)	Weight Loss (gm) Retainer	Weight Loss (gm) Ball	Brg. Clearance (In.) Before	Brg. Clearance (In.) After	Remarks
225	RB-HP-15-1	165	20	0.012	0.010	0.0023	NC**	Stable operation
	Pressed Steel	144		ND**	ND	0.0006	NC	
450	RB-HP-15-1	180	25	0.020	0.002	0.0033	NC	Stable operation
	Pressed Steel	138		ND	ND	0.0007	NC	
450	RB-HP-15-1	185	24	0.014	0.001	0.0034	NC	Oil temp 6°F higher before shutoff at start of test
	Pressed Steel	145		ND	ND	0.0008	NC	
225	WRP-140-2	144	20	0.183	0.001	0.0025	NC	Wt. loss of retainer included approx. 0.17 to 0.18 gm of oil
	Pressed Steel	137		ND	ND	0.0013	NC	
450	WRP-140-2	167	24	0.212	0.001	0.0025	NC	Wt. loss of retainer included approx. 0.16 to 0.19 gm of oil
	Pressed Steel	140		ND	ND	0.0015	NC	

*Total static torque for both bearings
**Not determined or no change

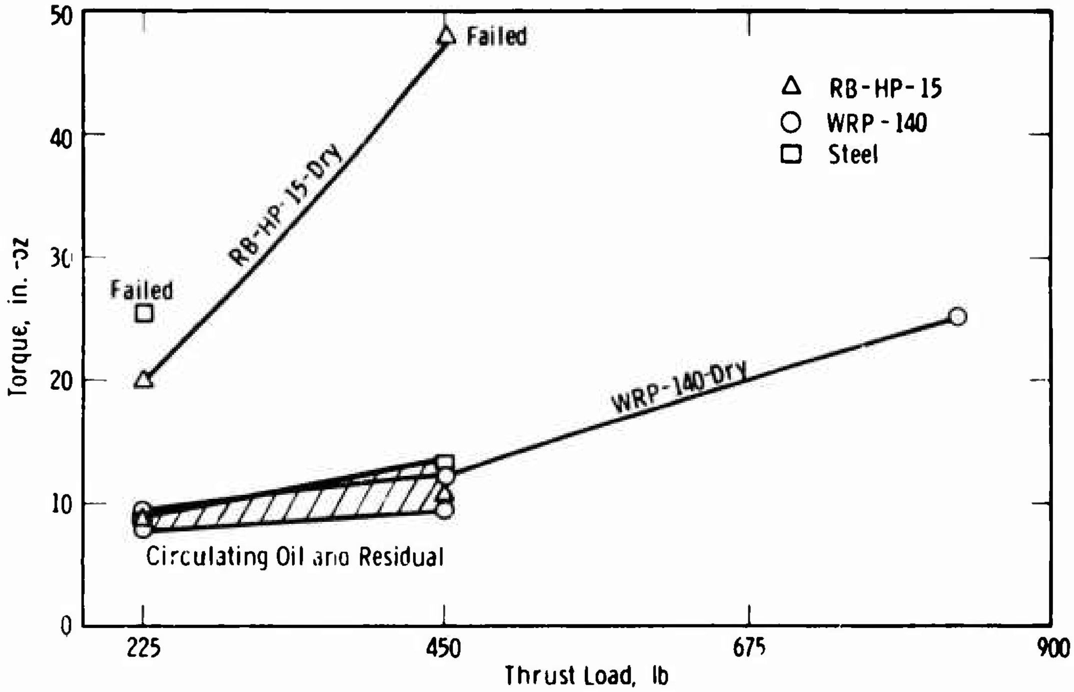


Temperature as a Function of Load for Steel and Various Solid Composite Retainers in 206 Ball Bearings for (1) Circulating Oil, (2) Residual, and (3) Dry Operation

Figure 2. Temperature vs Load For Solid Lubricated Bearings

The bearing torque and operating temperature for the various loads increased over that of the corresponding circulating oil operation. Although using temperature was not the most sensitive technique for determining the change, temperature provided the most information in this series of tests, as noted in Figure 2. The temperature increase of the bearing with the WRP-140-2 retainer was 20% higher than that of the circulating oil tests at the 450-lb load. The temperature increase for the operating bearing with the RB-HP-15-1 retainer was 20% at 225-lb and 35% at 450-lb over those obtained with the circulating oil system.

In the residual oil tests, when the load was doubled, the operating temperature increased approximately 13%. This increase was due in part to the loss of the cooling effect obtained when the oil was being circulated. Without the oil, the bearings stabilized at the higher temperatures after approximately 10 to 12 min. It was estimated that the oil in the bearing system drained completely within 1 to 2 min after the oil flow was terminated.



Torque as a Function of Load for Steel and Various Solid Composite Retainers in 206 Ball Bearings for (1) Circulating Oil, (2) Residual, and (3) Dry Operation

Figure 3. Torque vs Load For Solid Lubricated Bearings

The oil temperature of the facility bearing varied over a range of 6°F during the various tests and may have been influenced by the different operating temperatures of the test bearing.

The change in static torque values was small and within the range observed for the circulating oil tests, as noted in Figure 3. This was significant and indicated that bearing operation was stable and would level off at a higher operating temperature. This series of tests was run with no flow of cooling air over the bearing or housing structure. If cooling air had been used, the stabilized temperature would have been correspondingly lower.

Dry Operation Tests

The dry operation tests were made under extremely severe conditions. The bearings were wiped dry of oil before test, were started dry, and were run without oil for 0.5 hr at approximately 14,000 rpm. The two candidate retainer materials were evaluated in bearings at thrust loads of 225 lb and 450 lb. The WRP-140 retainers were also operated at a load of 800 lb for 1.5 hr without failure. Both bearings in the tester were used as test bearings.

TABLE VIII

DRY BEARING OPERATION

Speed: 13800-14000 rpm
Time: 0.5 hr

Load(Lb)	Bearing Retainer	Bearing Temp. (°F)	Static* Torque (In.-Oz.)	Weight Loss (gm) Retainer	Ball	Brg. Clearance (In.) Before	After	Remarks
225	RB-HP-15-1	180	30	0.039	0.002	0.0033	NC**	Stable operation
	WRP-140-1	151		0.002	0.001	0.0036	NC	
225	WRP-140-1	149	18	0.004	0.001	0.0035	NC	Stable operation
	WRP-140-2	147		0.008	0.001	0.0022	NC	
450	RB-HP-15-1	257	61	Failed	0.001	0.0025	NC	Retainer RB-HP-15-1 broke at end of test
	WRP-140-1	190		0.002	0.001	0.0044	NC	
450	WRP-140-1	169	24	0.009	0.001	0.0041	NC	Stable operation
	WRP-140-2	172		0.008	0.001	0.0020	NC	
800***	WRP-140-1	205	52	0.006	0.002	0.0025	NC	Stable operation for extended time
	WRP-140-2	211		0.215	0.003	0.0034	NC	

*Total static torque for both bearings
**No change
***Bearing operated for 1.5 hr and was satisfactory at shutdown

The results of tests are shown in Table VIII. Again, using temperature was a better method of evaluating bearing and retainer performance than using bearing torque, as noted in Figure 2 and Figure 3.

Satisfactory bearing operation was obtained at a load of 225 lb when either the WRP-140-1 or the RB-HP-15-1 retainer materials were used. The increase in temperature of the WRP-140-1 bearing was approximately the same in the residual tests. However, the RB-HP-15-1 bearing operated at 180°F. This temperature was similar as that for the 450-lb load in the residual oil tests and was 30°F higher than that for the WRP-140-1 bearing operating under the same conditions.

At the load of 450-lb, the RB-HP-15-1 bearing failed (29 min) just before completing the 0.5-hr test. The temperature had stabilized at approximately 260°F. The retainer split through the ball pockets; then the rings broke into smaller segments, as noted in Figure 4. The test unit

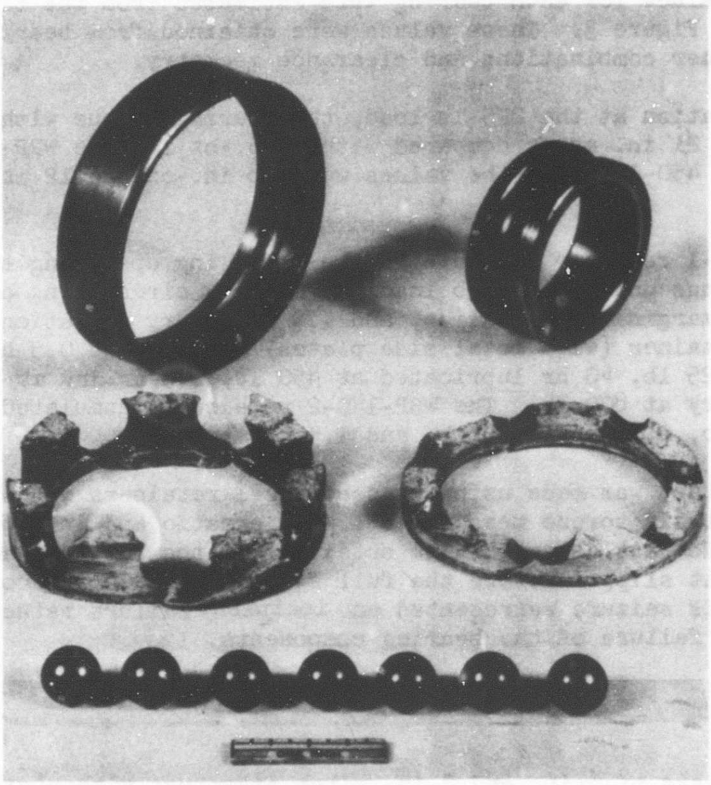


Figure 4. Ball Bearing With Failed RB-HP-15-1 Retainer

was immediately shut down. The bearing on which the retainer had failed was in excellent condition and had been satisfactorily lubricated until

failure occurred. No further testing was accomplished with the RB-HP-15-1 retainer material. The bearing may have failed because the retainer expanded and decreased the clearance between the inside diameter of the retainer and guide lands of the inner race. It is believed that the bearing would have operated satisfactorily in the range of 210 to 220°F had not failure of the retainer occurred. The rear bearings using either the WRP-140-1 or the WRP-140-2 retainers operated at approximately 20°F higher than the front test bearing in similar tests because of the heat "soak" through the shaft and support structure.

The stable temperature for the bearings with WRP-140-1 or WRP-140-2 retainers was 170°F at the 450-lb load. The temperature appeared to increase almost linearly from 170 to 210°F for operation up to 800 lb. A significant difference in retainer weight loss was noted for the two test bearings. This difference was due to the amount of oil lost in the one bearing. It was believed that one bearing was running under more duress and that the oil loss from the retainer was greater. The individual torque values for each bearing were estimated from the test data and are shown in Figure 3. These values were obtained from bearings with random retainer combinations and clearance geometry.

For dry operation at the 225-lb load, the bearing torque with a RB-HP-15-1 retainer was 21 in.-oz as compared with 9 in.-oz for the WRP-140-1 retainer. For the 450-lb load, the values were 48 in.-oz and 12 in.-oz respectively.

The RB-HP-15-1 retainer accumulated the following operating time in different bearings under a 225-lb load: 40 hr in a circulating oil system, 1.5 hr with marginal lubrication, and 1.5 hr of dry operation. The WRP-140-1 retainer (with metal side plates) accumulated 0.5 hr residual and dry at 225 lb, 40 hr lubricated at 450 lb, 0.5 hr dry at 450 lb, and 1.5 hr dry at 800 lb. The WRP-140-2 retainer accumulated 1.0 hr dry at 225 lb, 0.5 hr at 450 lb, and 1.5 hr at 800 lb.

One cursory test was made using pressed steel retainers without lubrication. The static torque measurements were erratic and in the range of 25 in.-oz. No test data could be obtained for the bearings seized and the drive belt slipped before the full speed of 14,000 rpm could be reached. This seizure represented an incipient failure rather than catastrophic failure of the bearing components.

GEAR TESTS

Steel spur gears of 6-in. and 2-in. pitch diameters with a diametral pitch of 12 were evaluated in the functional tests; 2.5-in. pitch diameter solid lubricating idlers of WRP-140 and RB-HP-15 were used. The configuration of the idler gears was similar in tooth profile to that of the large load gear except the root was undercut to provide additional tooth tip clearance of the load gears. Four idler gears

were used. The suffixal letter after the material designation identifies as follows: WRP-140-A, WRP-140-B, RB-HP-15-C, RB-HP-D. Each of the idler gears incorporated a ball bearing with a WRP-140 retainer in the steel gear hub. The WRP-140-A and WRP-140-B gears had a special oil reservoir in base of idler material; and a corresponding reservoir in the outer edge of the steel hub. A set of test gears along with the idler gear components, including the hub and bearing, is shown in Figure 5.

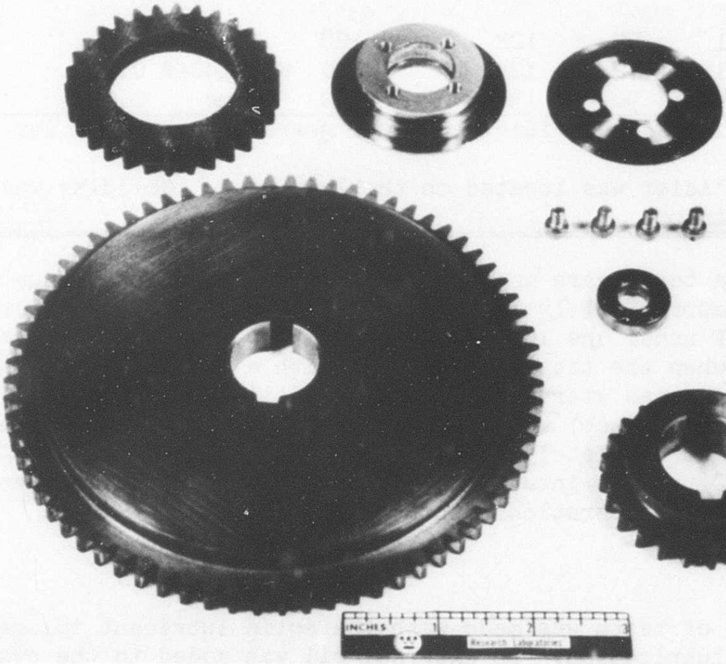


Figure 5. Steel Load Gears and WRP-140-A Idler Gear Components

The gear train was operated in a "four-square" gear test apparatus under various loads at pinion speed of 14,000 rpm. A detailed description of the apparatus, test gears, and test procedure is given in the section entitled "Experimental Program". The gears were evaluated under two major operating conditions to simulate both normal and failed lubricating systems in the transmission.

<u>Circulating Oil</u>

The first series of tests was made with and without the solid lubricating idlers in a normal oil-lubricated system.

The 40-hr duration tests were made at tooth loads of approximately 160 lb [640 lb (ppi) for each in. of tooth width] and 320 lb (1280 ppi). The results of tests are shown in Table IX.

TABLE IX

CIRCULATING OIL GEAR OPERATION

Speed: 13500-14000 rpm of pinion
Oil Flow: 600 cc/min
Time: 40 hr
Tooth Width: 0.250 in.

Tooth* Load (Lb)	Torque: Static (In.-Lb)	Torque: Dynamic (In.-Lb)	Temperature (°F): Oil	Temperature (°F): Gear	Idler** Material	Weight Loss (gm): Pinion	Weight Loss (gm): Idler
160	12.1	72	122	128	None	0.001	None
160	12.1	72	124	130	WRP-140-A	0.001	0.036
320	17.4	96	134	150	None	0.003	None

*Stable operation of both idler and load gears was observed for all three loads.

**The WRP-140-A idler was located on the load gear. No idler was used on the load pinion.

Operation of the test gears was satisfactory over the load range with an oil flow of approximately 600 cc/min. The average gear temperature was 128 to 130°F under the 160-lb load condition. No difference in operation was noted when the test was repeated with a WRP-140 idler in the system. The idler was started with absorbed oil in the material, but the reservoir (in the hub) was not filled. After tests, the reservoir was observed to be completely full. Another run was made at 320-lb tooth load (1280 ppi) to insure that the teeth had sufficient fatigue strength for similar operation without lubrication.

Dry Operation

The last series of tests was made with the solid lubricant idlers as the only method of lubrication. No external oil was added to the system. The results of the 0.5-hr tests are shown in Table X. The gears operated satisfactorily and successfully completed the run without measurable wear on the tooth face. The gears were capable of extended operation. The only noticeable change from that of the lubricated tests was the significant increase in noise level. The gears were operated at both the 160-lb (640 ppi) and 305-lb (1220 ppi) load.

The temperature data and observed noise level indicated that somewhat better lubrication was obtained with the RB-HP-15 material than with the oil filled WRP-140 material. However, the wear rate of the RB-HP-15 idler was much greater and may not be satisfactory for long-time operation. A more significant improvement was made when the WRP-140 idler was meshed with the pinion rather than the gear. The addition of an idler on both the gear and pinion also improved lubrication. A comparison was made of the oil-lubricated gear teeth with those of the dry operation at the 1220-ppi load, in Figure 6. The difference in wear is slight. A polished wear area was noted at the dedendum of the gear teeth.

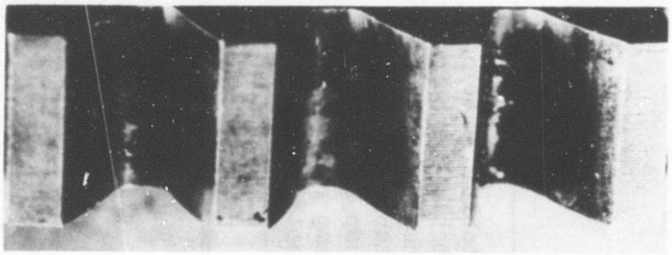
TABLE X

DRY GEAR OPERATION

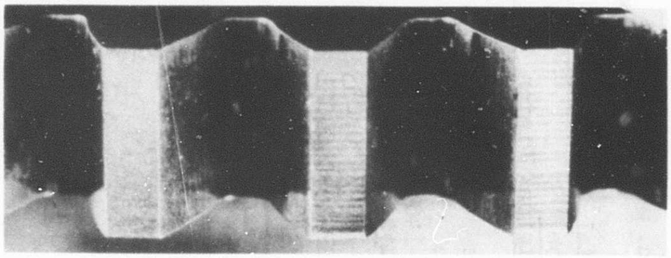
Speed: 12500-14000 rpm of pinion
Time: 0.5 hr
Tooth Width: 0.250 in.

Tooth Load (Lb.)	Torque Static (In.-Lb.)	Torque Dynamic (In.-Lb.)	Gear Temp. (°F)	Idler Material and Location Gear	Idler Material and Location Pinion	Weight Loss (gm.) Pinion	Weight Loss (gm.) Gear Idler	Weight Loss (gm.) Pinion Idler	Remarks
160	12.2	72	195	WRP-140-A	None	0.033	0.042	-	Stable operation
160	12.2	72	190	RB-HP-15-C	None	0.031	0.418	0.010	Stable operation
320	17.8	108	308	WRP-140-A	WRP-140-B	0.032	0.010	0.010	Temp range 308°F-280°F
170	12.3	72	175	None	WRP-140-A	0.013	-	-	Stable operation
305	17.8	108	300	WRP-140-A	WRP-140-B	0.033	0.0241	0.0107	Temp range 340°F-270°F
305	17.8	108	276	RB-HP-15-C	RB-HP-15-D	0.021	0.563	0.909	Temp range 302°F-260°F
360	17.9	108	320	None	WRP-140-A	ND*	ND	ND	Tooth failure**

*Not Determined
**One tooth of 6-in. gear broke after 23 min



Circulating Oil Operation

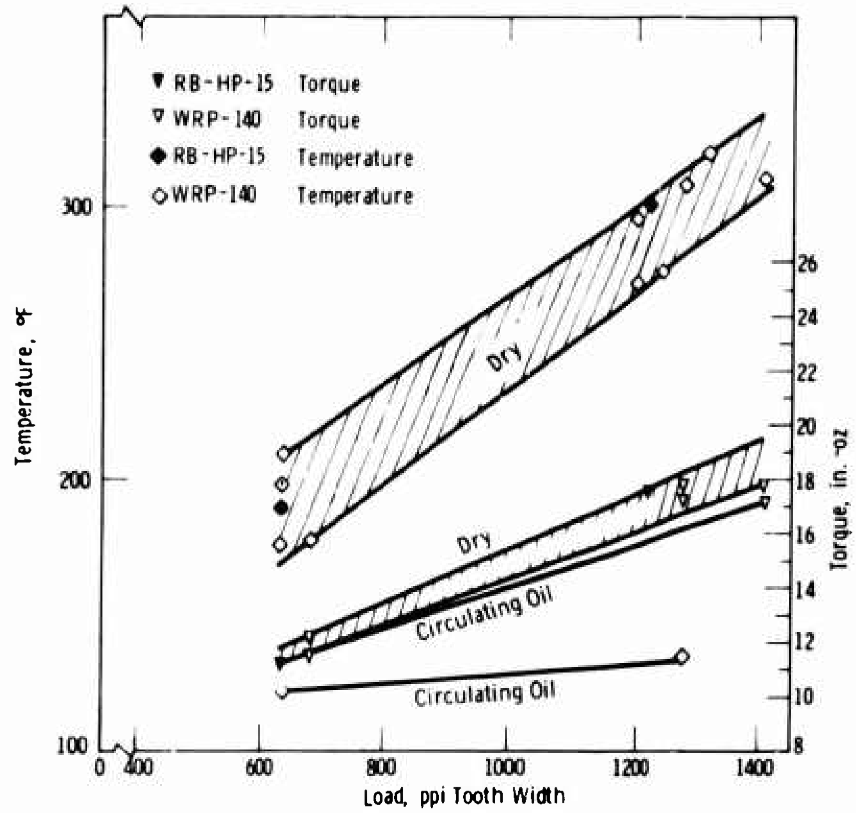


Dry Operation

Figure 6. Gears After 0.5 Hr Of Operation At 1220 ppi

A comparison of temperature and torque for the various runs is shown in Figure 7. A greater difference in temperature is noted for the tests at the two loads as compared to the torque values. This indicated, even though the temperature was obtained with a pyrometer in the static condition, that it was superior to measuring torque. The static and dynamic torque values were apparently responsive to other components in the system.

The dynamic torque readings were used to indicate a severe change in operation and did indicate a gear tooth failure on one of the special tests. The test was made at an exceptionally high tooth load of 360 lb (1440 ppi) at 12,500 rpm; a fatigue failure of one 6-in. gear tooth occurred after 23 min of operation. The idler gear was not damaged but started to bounce as the pinion teeth passed over the missing gear tooth. Considerable scoring and pounding were noted on several of the pinion teeth. The unit was still operating when shut down, but approximately 40% of the load torque had been lost.



Temperature and Torque as a Function of Load for 6-in. Gears with and without Idlers for (1) Circulating Oil, and (2) Dry Operation

Figure 7. Temperature and Torque vs Load For Solid Lubricated Gears

SOLID LUBRICANT COMPATIBILITY WITH MIL-L-7808 OIL

Lubrication tests of the immersed solid lubricants and an infrared (IR) spectrographic analysis of the oil were made to insure compatibility of the two materials when used together in a helicopter transmission. The lubricating tests were made using the immersed specimens from the residual oil screening tests.

The oil, after the 45-day immersion tests at 150°F, was analyzed to determine if any chemical change had occurred to affect the acidity or other corrosion characteristics of the oil. A visual comparison of the oils was first made. A significant difference in color was noted for two of the materials. The two oil samples which contained the RB-CP-1 and RBP materials appear to have darkened and may not be satisfactory. The most desirable WRP-140 material, did not show any discoloration and was considered to be satisfactory.

In addition, previous immersion tests with petroleum oils, as accomplished in related development programs, indicated that petroleum oils were also inert to WRP-140.

A complete IR analysis was made on an unused oil sample and on one that contained the WRP-140. The results are shown in Figure 8. Within the limits of IR sensitivity the two oils appear to be chemically identical.

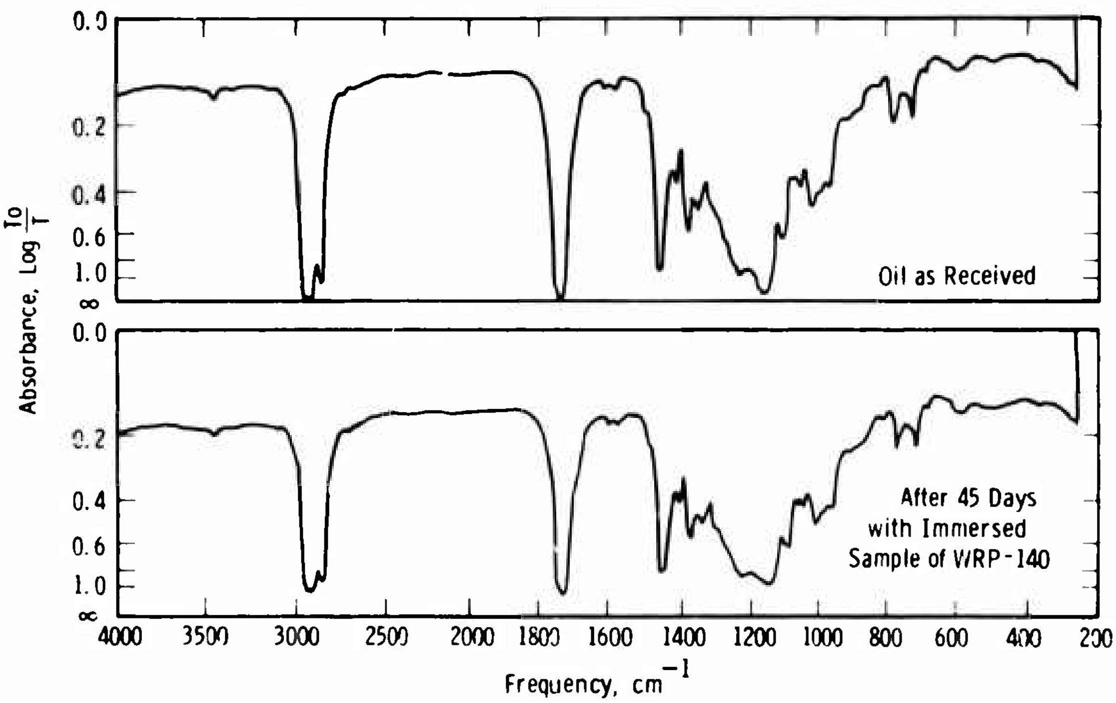


Figure 8. Infrared Spectrum Of MIL-L-7808 Oil

The peak at a frequency of 1170 cm^{-1} is most likely due to a (-C-O-C-) linkage, is somethat more intense in the as-received oil than in the other; however, if any appreciable hydrolysis of the esters present had taken place, an alcohol would have been formed, and enhanced absorption at a frequency of 333 cm^{-1} should have been observed. The slight difference in the peaks at 1170 cm^{-1} could be explained on the basis of a slight difference in sample size or in operating parameters of the instrument.

DISCUSSION OF RESULTS

Solid lubricants in the form of graphite and molybdenite ore were known for centuries. However, they did not come into prominence until World War II when the military services used these materials in lubricating systems to prevent seizure of metal surfaces where oils and greases were unsatisfactory. Significant recent advances are noted. Bowen,[2] Devine,[3] Boes and Bowen,[4] Bonis,[5] Campbell and Van Wyk,[6] and Boes[7] have developed a variety of composite materials satisfactory for lightly loaded bearing applications. Bowen[8] has shown that composite materials could be used to lubricate both gears and bearings under heavy loads. Additional information regarding lubrication and other subjects related to helicopter transmissions is found in Appendix II.

During this program, powder lubrication was not considered because of the current work of Wallerstein[9] in lubricating gears and bearings of a jet engine. Instead, the upper range of loads on the bearings and gears of this helicopter transmission study was increased twofold to obtain a wider spectrum of operation.

The bearing and gear configurations used in transmissions are dependent on the type of load along with the rotating speed of the particular component in the system. Conrad-type bearings rather than angular contact bearings were used, even though thrust loads were used in the tests. In general, the Conrad-type bearings exhibit approximately 90% of the rated thrust load values of those for angular contact bearings. The gears employed short addendum teeth on the gear and long addendum teeth on the pinion to improve the bending strength of the gear. This tooth profile increased sliding action of tooth tip by approximately 20% over that of a conventional tooth profile. The bearings and gears were of aircraft quality and were made from conventional materials.

Both the bending stress and compressive (Hertz) stress were calculated for the gears. Considerable variation can be achieved in selecting the service factors involved in these calculations, as noted in Appendix I.

During the preliminary bearing tests with the circulating oil system, several conventional design bearings with pressed steel cages failed when the oil flow was less than approximately 350 cc/min. An examination of the inner ring of a failed bearing revealed that the race was a light blue color. The ball path of this part was a dark blue color. The Rockwell hardness was measured at 58.0 R_c. The ball path was located high on one shoulder. Severe denting was found within this ball path along with light denting found in the bottom of the raceway. The outer race was brown in color with a hardness of 55.0 R_c. The ball path was very wide, pitted, and dented. The ball path was located in the bottom of the race and indicated that possibly it was running against the counterbore shoulder. The counterbore was badly dented. The balls found in

this bearing were burnt and contained large, flat skid marks. ^ the balls were grooved from riding against the shoulder. The balls measured a hardness of 53.0 R_c.

It is believed that temperatures of 600°F were experienced on the balls and approximately 450°F in the outer race. The inner race does not appear to have experienced the high temperatures of the balls or the outer race. This analysis indicated that a bearing using higher shoulders would be desirable to increase the reliability of operation under conditions of marginal lubrication.

The variables in the gear tests were held to a minimum. The design of the tester and gears eliminates many of the service factors experienced in normal gear operation. In addition, diligence was exercised in obtaining both load torque and driving torque measurements. The driving or friction torque was monitored continuously and recorded intermittently by using the special torque telemetering system. Although this combined torque of the driving belt and tester was higher than the test gear torque, it was constant and was calibrated prior to each test. In a typical test, such as the 800 lb dry run, and driving the 6-in. gear using two WRP-140 idlers, the torque peaked during the first 0.3 sec of the starting cycle and leveled off at 108 in.-lb. The friction torque was recorded at 5 min intervals. No deviation from the 108 in.-lb value was noted.

The load torque values were obtained using the SR-4 strain indicator (1) prior to test, (2) several minutes after a prerun, and (3) again at the end of the test. The greatest load torque loss occurred at the 320-lb (1280 ppi) load in which approximately 8.5% of the load was lost. No significant or visible wear had occurred on the teeth. At the rated load torque of 160 lb, 4.0% of the load was lost.

The test results indicated that both the WRP-140 and RB-HP-15 materials exhibited desirable characteristics for the solid lubrication of high-speed, high-load gears and bearings. The WRP-140 material was the superior material. If the RB-HP-15 is to be used in bearings, the maximum load under dry conditions would be limited. If used in gears, the wear life would be limited even though the idler gear operated in the circulating oil condition. The RB-HP-15 idler gear would provide satisfactory emergency operation of the transmission if the system were so designed that the idler would be brought into mesh at the time of lubrication system failure.

SCREENING TESTS

Equipment

The screening tests were performed with the wear and friction test apparatus shown in diagram of Figure 9. The test specimen was held in the left side of two shoes and rubbed against the edge of a rotating hardened 440-C

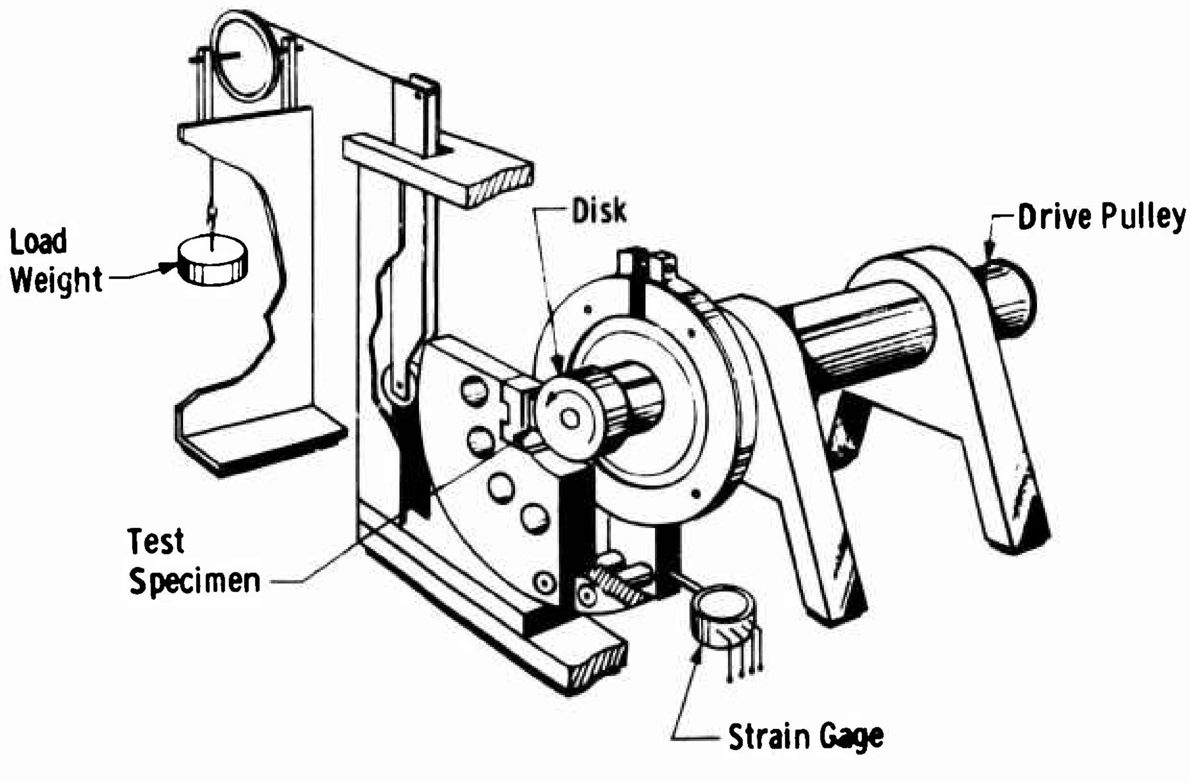


Figure 9. Wear and Friction Test Apparatus

stainless steel disk (1.375-in. in diameter) and shaft assembly. The speed of the disk was 7100 rpm to provide a sliding velocity of 2560 ft/min. The shoes were mounted on a torque arm and bearing assembly that pivoted in a concentric arc about the disk shaft. The load was applied to the test specimen through the left shoe with a lever and dead weight.

The screening tests were made on the same specimen under various loads for each of the three operating conditions (circulating oil, residual or marginal, and dry.) The tests were made in a random order to minimize any variables encountered in the procedure or environmental conditions.

Wear, friction values, and temperature were observed on the test specimens. The wear was measured as the chord of the arc worn on the 0.25-in. face. Since the wear area varied for the different specimens, the unit force varied. The results of tests were observed as scar width in mm and reported as wear for various loads in lb per sq-in. The friction was determined at 5-min intervals by observing the strain gage deflection on an SR-4 strain indicator as transmitted from a dynamometer ring. The temperature was obtained by using a thermocouple located in the test specimen approximately 0.030 in. behind the rubbing surface. The temperature values always lagged behind the torque values during the first 5 to 15 min of the test. The average temperature reported generally reflected the value obtained after it had stabilized.

Test Procedure

General

The specimens were machine-finished to a dimension of 0.25 in. x 0.75 in. x 0.50 in. The thin film specimens were coated on the 0.25 in. x 0.75 in. faces. Thermocouple holes of 0.030-in. diameter were drilled 0.375 in. deep, parallel to the 0.75-in. rubbing surface, and approximately 0.030 in. behind the rubbing surface. This insured a central thermocouple location for each of the two runs made on a virgin area of each face. All of the specimens were weighed and placed in a desiccator until ready for test.

The duration of each test was 30 min. The load on the specimens was applied by a dead weight system in the running condition prior to test and released in the running condition after test. Generally, two tests were made on each specimen face.

Circulating Oil Operation

MIL-L-7808 oil was used as the lubricant for the solid composite lubricant specimens. The oil was wick-fed on the rubbing surface of the disk at an approximate rate of 10 drop/min. Although the oil was applied on the surface of the disk, no provisions were made to collect the oil and return it to the reservoir. The procedure for obtaining the data during test along with the information after test was the same as that described above. Series of tests were made at 1500-psi load and again at 3000-psi load.

Residual or Marginal Operation

The residual or marginal lubrication tests were made on the specimens after they had been removed from the oil compatibility immersion tests. As the specimens were removed from the oil, they were wiped dry, the oil was drained from the thermocouple holes, and the specimens were weighed and subjected immediately to the wear and friction test. No additional lubricant was used in the system. The test loads were in the range of 150 to 200 psi.

Dry Operation

The specimens were brushed, rinsed in anhydrous ethyl alcohol, and dried immediately prior to test to remove all traces of contamination due to handling. The disk was also cleaned with an abrasive and was polished and rinsed in anhydrous ethyl alcohol prior to test. The test and inspection procedures were the same as outlined above. The loads on the specimens were the same as those of the marginal lubrication tests.

SOLID LUBRICANT COMPATIBILITY WITH MIL-L-7808 OIL

The candidate solid lubricant specimens were immersed in separate 250 cc breakers containing 50 cc of MIL-L-7808 oil at 150°F for a period of 45 days. The specimens were removed after 45 days for a weight determination and evaluation of lubricating properties (in the screening). Any change in the physical-chemical characteristics of the fluid was noted by visual observation and by comparison of infrared spectrums before and after the specimen immersion.

BALL BEARING LUBRICATION

Equipment

The apparatus was used to evaluate modified bearings with solid lubricants under conditions of circulating oil operation, residual operation, or dry operation. The apparatus consisted of the test bearing system and loading device, drive motor, oil reservoir, heat exchangers, filter, sight port, fluid circulating system, and flow meters with flow controls. A photograph of the component arrangement of the system is shown in Figure 10.

The bearing tester used two test bearings mounted on a shaft and driven by a pulley located near the rear bearing. The front bearing was mounted in a floating housing which permitted the applied thrust load to be distributed evenly to both bearings. The load was maintained through a hydraulic ram located on the front of the apparatus.

Shields were used on the inside of each bearing housing to prevent transfer of oil or mist from one bearing to the other. In addition, the shield was used as an integral part of the lubricating system for each bearing. The shield was double-walled and formed an annulus to supply a sufficient jet stream of oil through three equally spaced 0.020-in.-diameter holes to the bearing. Each of the three streams was directed axially into the bearing between the outer diameter of the retainer and the inside diameter of the outer race. An oil return channel was located at the bottom of the housing to drain the oil from the bearing. The oil was collected in an oil groove in the bottom of the frame, drained to the center sump, and returned through a gravity drain line to the reservoir.

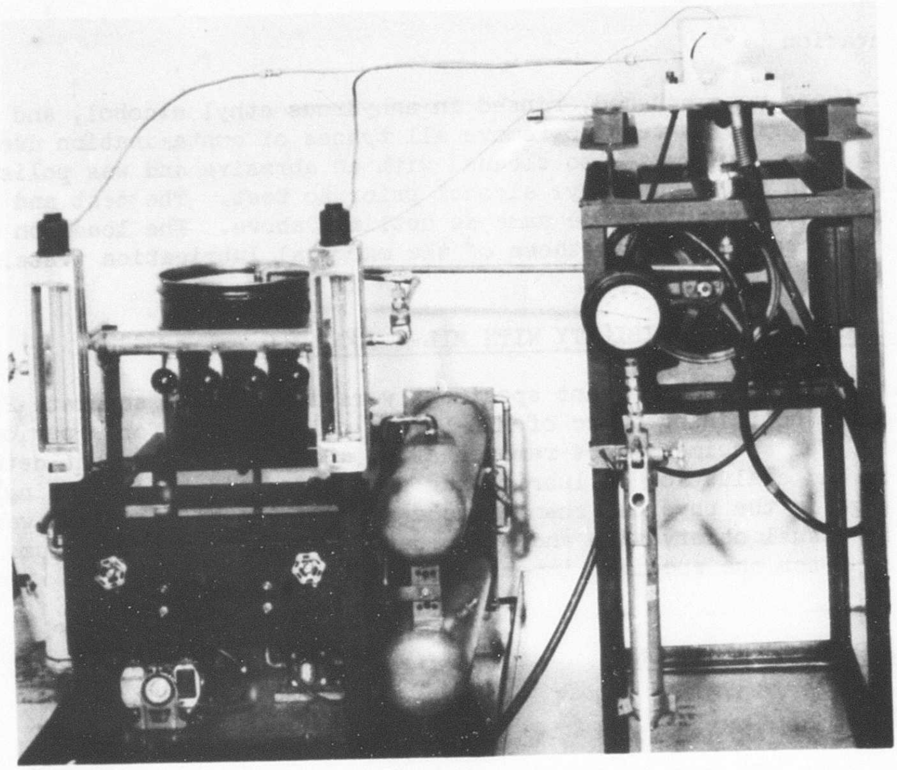


Figure 10. Bearing Test Apparatus

The oil was supplied to the bearing by a positive displacement gear pump through a filter, sight port, pressure manifold and flow meter with a throttling valve ahead of the meter. The manifold pressure was maintained at 50 psi by means of a pressure relief valve and bypass system of the gear pump. A heat exchanger was located between the reservoir and pump to cool the oil. No effort was made to determine the heat transfer from the oil. An instrument console was used to record bearing and oil temperatures; it served as a guide to any impending bearing failure.

The test apparatus was belt-driven by a 7-1/2 hp, 3400 rpm, a.c. motor. Special high-speed precision belts and pulleys were used to minimize vibration. The bearings rotated at speeds in the range of 13,800 rpm to 14,000 rpm. A stroboscope was used to determine the operating speed of the bearings for each test. The rated load for each bearing was a thrust load of 450 lb.

Friction torque measurements were made on the loaded bearings in the static condition both prior to and after test. A cantilever arm and a deadweight system were used on the drive shaft to determine static torque. With the drive belt removed and the balanced arm installed in a horizontal position on the drive shaft, weights were added until the shaft just moved.

Test Bearings

Aircraft quality class 5AFBMA 206-size, 30 mm bore, deep-groove, Conrad-type ball bearings were modified to use the one-piece solid lubricant retainers. The internal radial clearance was increased from 0.00085 to 0.0040 in. One land of the outer race was ground to permit assembly of the balls and retainer.

Test Procedure

The retainer materials were fabricated, inspected, cleaned with anhydrous ethyl alcohol, identified, and stored in a desiccator until ready for test. The bearings were modified, identified, oiled, greased, and placed in individual containers until ready for test.

The pretest procedure consisted of cleaning and weighing the bearing components (weighing the retainer and assembly). The bearings were installed in the tester with the ground lands facing inward. The top and bottom position and the match of inner and outer races were recorded for each test bearing as assembled in the housings. The oil supply lines and loading device were installed, and the unit was subjected to a check run of several minutes to insure satisfactory operation.

Circulating Oil Operation

The oil flow to each bearing was first determined for bearings with pressed steel retainers. At low oil flows and a rated load of 450 lb, the bearings failed after7 to 20 hr of operation. It was necessary to increase the flow to each bearing from the initial value of 250 to 400 cc/min. All subsequent circulating oil bearing tests with solid composite retainers were made by using the flow rate of 400 cc/min. The bearings with pressed steel retainers were operated for 40 hr along with those using solid composite retainers to obtain comparative information on the composite retainers for normal lubricated conditions.

The procedure at start-up of the bearing lubrication tests was similar to that of actual bearing operation in a transmission. The load was applied on the bearings in the static condition. The drive motor and the related oil circulating and cooling systems were started at the same time. The unit was operated for 0.5 hr before the 40-hr duration tests were started. During the pretest period, the oil temperatures, pressures, and flow rates were monitored to establish stable operation.

After test, the drive and lubricating systems were shut down, and the bearings were removed from the test unit. Regular laboratory procedures were used to inspect and weigh the bearing components. All of the circulating oil tests were made at the rated thrust load of 450 lb on each bearing.

Residual or Marginal Operation

One test bearing and one facility bearing were used in this series of tests. The facility bearing incorporated a pressed steel retainer similar to that used in the base line tests. The bearings were assembled in the test apparatus and given a 0.5-hr precheck operation similar to the oil circulating tests. Oil flow and bearing temperatures were monitored during the precheck operation.

The residual or marginal lubrication test was accomplished by shutting off the oil supply to the test bearing after the precheck operation and by operating 0.5 hr on the residual oil supply. The test structure utilized a number of ports between the bearing housing and frame to provide sufficient ventilation and to prevent vapor transfer to the test bearing. The ports also provided a method of observing the amount of vapor that formed in the test bearing housing.

Dry Operation

Two test bearings were used for the dry operation tests. The candidate solid lubricant retainers were given a vacuum impregnation in oil and allowed to soak for an additional 2 hr. The retainers were wiped dry after being removed from the oil, weighed, and assembled in the previously cleaned bearings. The bearing tester frame, housings, shaft and shields were thoroughly degreased before the bearings were installed for test. The oil circulating system was not connected to the tester.

Static torque readings were obtained on the loaded bearings prior to test. The tester was then started and operated for 0.5 hr. The unit was stopped, and the bearings were removed and subjected to the normal post-test inspection. Repeat tests were made at various loads on new bearings.

GEAR LUBRICATION

Equipment

The spur gear solid lubrication tests were made on a "four-square" test apparatus which used a closed power circuit principle. The apparatus utilized a 3:1 gear ratio with a pinion speed of 14,000 rpm.

It consisted of two mounted pedestals with gears connecting two parallel shafts, as noted in Figure 11. The helical facility gears were located in the left pedestal, or gearbox. The test gears were located on the cantilever end of the shafts external to the right pedestal. A cover plate was used over the test gears when operated with the oil circulating system. Two facility bearings were used on each shaft in each pedestal to provide accurate alignment of the gear. Special high-speed shaft seals

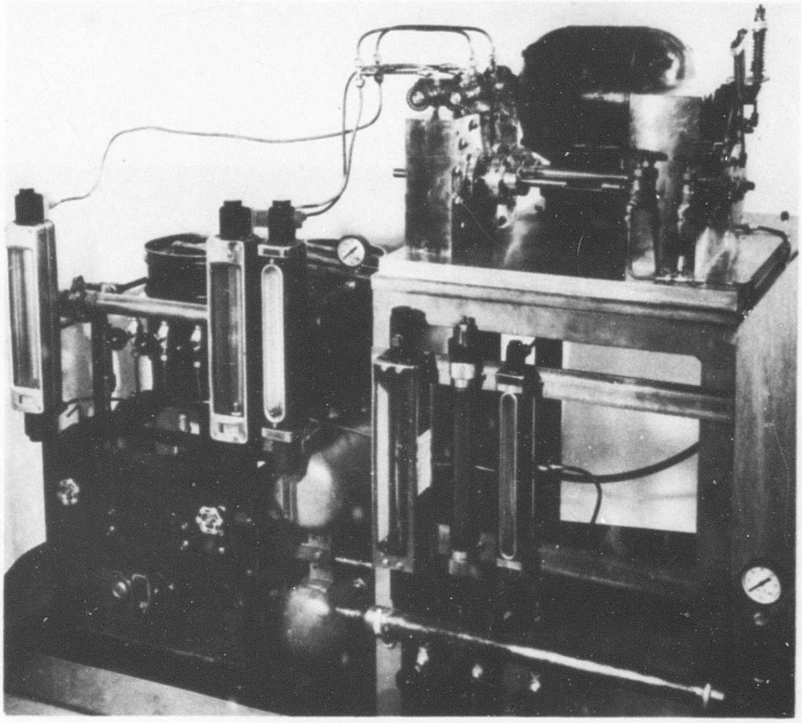


Figure 11. Gear Test Apparatus and Lubrication System

were used to retain the oil in the bearings. The left pedestal used spring-loaded "O" ring seals, and the right (test) pedestal incorporated magnet carbon face seals.

The unique feature of the tester was that high loads could be applied to the 0.25-in.-wide test gears, and only the power required to overcome friction in the system was necessary to rotate the gears. The load was applied to the torque coupler located on the smaller diameter shaft. Weldable strain gages were located on this same shaft to measure the load applied to the gears. All load measurements were made in the static condition prior to, during, and after test. The load was applied with a lever and force system.

The gear tester and drive motor with related gear and bearing lubricating system are shown in Figure 11. The gear test control and power torque monitoring system are shown in Figure 12. A block diagram of this related equipment is shown in Figure 13. Separate lubricating systems were used for the facility components and the test gears. Each system contained a heat exchanger, flow meters, gages, manual valves, and other related equipment. Oil flow to each set of facility bearings and gear set was individually measured and controlled. Flow in the separate system for the test gears was also observed and controlled during test. Provisions

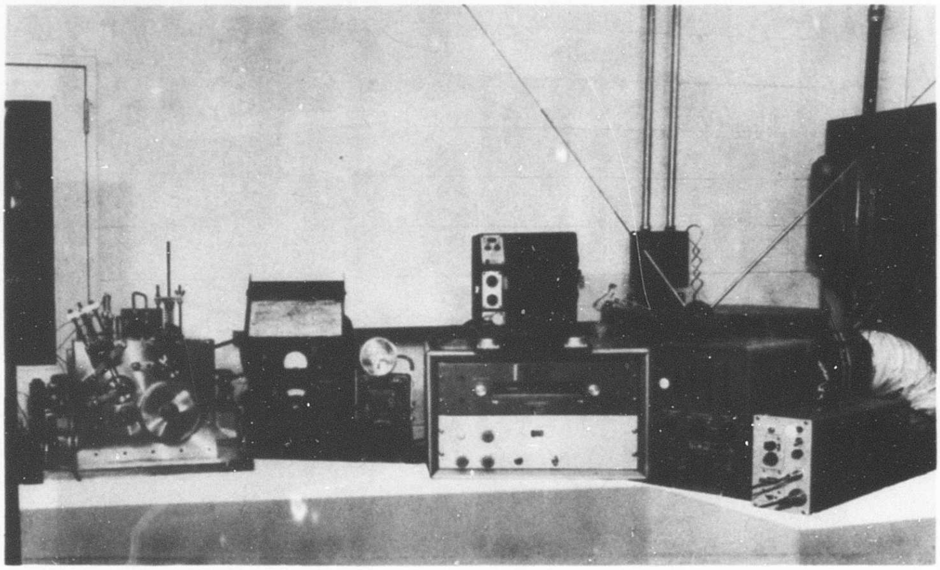


Figure 12. Gear Test Apparatus and Torque Receiver Instrumentation

were made to record the oil temperatures of the facility bearings and the gear train in the left pedestal along with the reservoir temperature. For the test gear oil system, provisions were made to record three temperatures: the oil at the entrance of the tooth face, at the exit of the tooth face, and in the reservoir.

The "four-square" test apparatus was belt-driven by a 7-1/2 hp, 3 phase, 3400 rpm, a.c. motor. Continuous torque of the system was obtained by an FM/FM transistorized telemetry system. Transducer strain gages were mounted on the motor shaft along with a D battery power source and two oscillators and a transmitting antenna. A receiving antenna, power supply, tuner, discriminator, and oscillographic recorder were located in the area adjacent to the test apparatus. The transmitter operated as follows:

1. The subcarrier oscillator supplied excitation voltage to the strain gage bridge.
2. As strain was picked up by the bridge, it was unbalanced; its output modulated the frequency of the subcarrier oscillator.
3. The frequency of the subcarrier oscillator modulated the frequency of the radio carrier, and the signal was transmitted by the transmitting antenna.

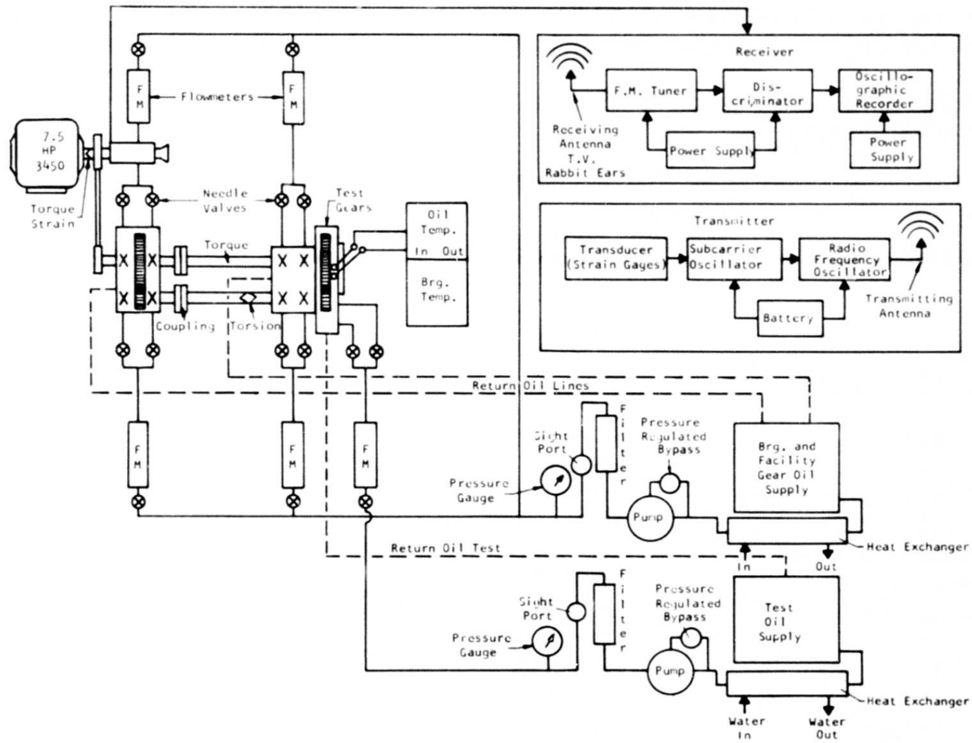


Figure 13. Oil Lubrication and Torque Monitoring System Diagram

4. A d.c. battery supplied the voltage for the transmitter.

The receiver and recorder operated as follows:

1. The signal was received via a receiving antenna (TV rabbit ears) and a tuner, which was tuned to the transmitting radio frequency.
2. The discriminator detected changes in the frequency of the subcarrier oscillator and converted these changes into d.c. voltage equivalents.
3. The d.c. voltage equivalents were recorded with an oscillograph.
4. The power supply furnished all the required power for the receiver, and the oscillograph has its own power.

Test Gears

The ground gears were fabricated from case hardening (Rockwell $60R_c$) AISI 9310 steel. The 12 diametral pitch, 0.250-in.-wide gears used short-long addendum teeth to provide maximum bending strength of both the gear and the pinion teeth. Each gear and pinion incorporated a proof diameter

flange 0.250 in. wide approximately 0.375 in. smaller than the pitch diameter. The pinion also included a groove extending in front of the flange to facilitate removal of the gear from the shaft. Summary design information of both the pinion and gear is noted in Table XI.

The idler gears were mounted on steel hubs with side plates, as previously noted in Figure 5.

TABLE XI

SUMMARY OF GEAR INFORMATION

Description	Load Gears Gear	Load Gears Pinion	Lubricating Gears Idler*
No. of teeth	72	24	30
Pitch dia., in.	6.00	2.00	2.50
Pressure angle, degree	20	20	20
Diametral pitch	12	12	12
Face width, in.	0.250	0.250	0.375
Addendum, in.	0.0537	0.1037	0.083
Dedendum, in.	0.1293	0.0793	0.102
Base circle dia, in.	5.63814	1.87938	2.3475
Outside dia, in.	6.1074	2.2074	2.6660
Rad of curvature at pitch point, in.	1.0260	0.3420	

*The length of the tooth was increased 0.025 in. by decreasing root diameter circle.

Test Procedure

Each idler gear, without the gear hub and hub bearing, was identified, inspected, cleaned with anhydrous alcohol, and stored in a desiccator until ready for test. The load gears, hubs and bearings were identified, oiled and placed in individual containers until ready for test.

As part of the pretest procedure, the load gears were cleaned, weighed, and installed on the gear shafts of the test head. The 6-in. gear was always selected as the driving gear. The lubricating idler gear was assembled and installed to engage with the load gear using a spring loaded floating shaft. The spring load for most of the tests was approximately 5 lb. A similarly loaded idler gear was used in several tests to directly lubricate the 2-in. load gear.

Circulating Oil Operation

The oil was fed to the test gears using two oil jets with a nozzle area of 0.015 sq-in. One jet was located in a vertical plane and directed downward with the oil stream lubricating both gears. A second jet was directed into the face of gears. Oil flow from each jet was approximately 400 cc/min. A cover over the test headwas used for all the oil tests and was designed to provide a dry pump for the test gears. A glass port was provided in the cover to permit visual inspection of the lubricating idlers, load gears, and oil nozzle spray patterns.

The torque load on the gears was always applied to the pinion shaft of the tester in the static condition. Usually a prerun was made for several minutes at partial load before the full load was applied. The full load was again checked after test and prior to disassembly.

Dry Gear Operation

Each solid lubricant idler gear (with hub and hub bearing) was impregnated with MIL-L-7808 oil in a vacuum and allowed to soak for an additional 2 hr. The gears were wiped dry after removal from the oil bath, weighed and installed on the floating shafts. New load gears were used for each test. A 2 min pretest full-load torque run was made in addition to the partial-load run as described in the circulating oil tests. The torque was checked again after 0.25 hr operation as well as at the end of each test.

The driving torque was monitored continuously during the test. Any change in operation of the gears could be detected with the transistorized telemetry drive torque instrumentation and also by the change of intensity of the audible noise.

CONCLUSIONS

It is concluded that:

1. Bearings with solid lubricant retainers and gears, using solid lubricant idlers, operated satisfactorily during normal oil lubrication and without external lubrication at high loads and speeds. This simulated emergency operation after lubrication failure in a helicopter transmission.

2. The life of the solid lubricants far exceeded the required 0.5-hr operation at a bearing thrust load of 800 lb and a gear load of 1200 ppi.

3. The life of the WRP-140 glass impregnated polyimide material was not dependent on surface wear but rather on the amount of oil retained in the integral reservoir.

4. The life of the RB-HP-15 silver alloy Teflon material was dependent on the surface transfer of the material to the mating metal surface. This life would be less than that of the WRP-140 material under continuous operating conditions.

5. The effect of load was more consistently correlated with the change in bearing or gear operating temperature than with torque. The temperature appeared to be a linear function of load and indicated that the WRP-140 material could be used for heavier loads.

REFERENCES

1. R. L. Johnson, et al., "Friction and Wear of Hot-Pressed Bearing Materials Containing Molybdenum Disulfide", NACA 2027, NASA Lewis Research Center, Cleveland, Ohio (1957).

2. M. J. Devine, E. R. Lamson, and J. H. Brown, "Inorganic Solid Film Lubricants", Journal of Chemical and Engineering Data, 6:79-82 (1961).

3. P. H. Bowen, "Dry Lubricated Bearings for Operation in a Vacuum", Transactions ASLE, 2:315-326 (1962).

4. D. J. Boes and P. H. Bowen, "Friction Characteristics of Self-Lubricated Composites Developed for Vacuum Service", Transactions ASLE, 6:192-200 (1963).

5. L. J. Bonis, "Technical Information on Standard Ilkoloy Family of Self-Lubricating Composite", Ilikon Paper AFIL-11, Ilikon Corporation, Natick, Massachusetts (1964).

6. W. E. Campbell and J. W. Van Wyk, "Lubricant Composite Materials", Lubrication Engineering, 20:12:463-469 (1964).

7. D. J. Boes, K. W. Grossett, and E. S. Bober, "Solid Film Lubrication Research", Part I, AFAPL-TR-66-110, Wright-Patterson AF Base, Ohio (1966).

8. P. H. Bowen, "Lubrication of Bearings and Gears for Aerospace Facilities", AEDC-TR-65-91, Arnold Air Force Station, Tennessee (1965).

9. S. Wallerstein, "Application of a Powder Lubrication System to a Gas Turbine Engine", Parts I and II, AFAPL-TR-65-43, Wright-Patterson AF Base, Ohio (1965).

DISTRIBUTION

US Army Materiel Command	4
US Army Aviation Materiel Command	6
United States Army, Pacific	1
Chief of R&D, DA	1
Director of Defense Research and Engineering	1
US Army R&D Group (Europe)	2
US Army Aviation Materiel Laboratories	18
US Army Limited War Laboratory	1
US Army Human Engineering Laboratories	1
US Army Ballistic Research Laboratories	2
Army Aeronautical Research Laboratory, Ames Research Center	1
US Army Research Office-Durham	1
Plastics Technical Evaluation Center	1
US Army Test and Evaluation Command	1
US Army Combat Developments Command, Fort Belvoir	2
US Army Combat Developments Command Transportation Agency	1
US Army War College	1
US Army Command and General Staff College	1
US Army Aviation School	1
US Army Tank-Automotive Center	2
US Army Armor and Engineer Board	1
US Army Aviation Test Activity, Edwards AFB	2
Air Force Flight Test Center, Edwards AFB	2
US Army Field Office, AFSC, Andrews AFB	1
Air Force Aero Propulsion Laboratory, Wright-Patterson AFB	1
Air Force Flight Dynamics Laboratory, Wright-Patterson AFB	1
Systems Engineering Group, Wright-Patterson AFB	4
Naval Air Systems Command, DN	15
Office of Naval Research	3
Naval Air Engineering Center, Philadelphia	1
Commandant of the Marine Corps	1
Marine Corps Liaison Officer, US Army Transportation School	1
Lewis Research Center, NASA	1
Manned Spacecraft Center, NASA	1
NASA Scientific and Technical Information Facility	2
NAFEC Library (FAA)	2
US Army Aviation Human Research Unit	1
US Army Board for Aviation Accident Research	1
Bureau of Safety, Civil Aeronautics Board	2
US Naval Aviation Safety Center, Norfolk	1
Federal Aviation Agency, Washington, DC	2
Civil Aeromedical Institute, FAA	2
US Army Medical R&D Command	1
The Surgeon General	1
US Government Printing Office	1
Defense Documentation Center	20

APPENDIX I

STRESS CALCULATIONS ON TEST GEARS

TOOTH LOAD

The power transmitted and the size of gears were selected as being representative of a normal gear train in a helicopter transmission of a limited horsepower rating. The power for a 1.00-in. wide gear was selected as 275 horsepower.

The desired speed was 14,000 rpm. A 2-in. pinion and a 6-in. gear were selected. The load on the gear tooth would be as follows:

$$W_t = \frac{33{,}000 \times hp \times 12}{2\pi r N}$$

where:

W_t = pinion tangential force in lb

hp = horsepower

r = pinion radius in in.

N = pinion rpm

$$W_t = \frac{33{,}000 \times 275 \times 12}{2\pi \times 1 \times 14{,}000} = 1240 \text{ lb}$$

The tooth load at the pitch point on the line of action is:

$$W_t = W_t \times \cos\theta$$

where:

θ = pitch angle of gears and was selected as 20°:

$$W_t = 1240 \times 0.9396 = 1200 \text{ lb/in. (ppi) tooth width}$$

The actual load on the 0.25-in.-wide gear was 300 lb.

BENDING STRESS

The bending stress is calculated as follows:

$$S_t = \text{(Load) (Tooth Size) (Stress Distribution)}$$

$$S_t = \frac{W_t K_o}{K_v} \cdot \frac{P_d}{F} \cdot \frac{K_s K_m}{J}$$

where:

S_t = Bending stress

W_t = Tangential load at pitch diameter = 300 lb

P_d = Diametral pitch of gear = 12

F = Face width = 0.25 in

K_o = Overload factor = 1

K_s = Size factor = 1

K_m = Load distribution factor = 1.5

K_v = Dynamic factor = 1

The geometry factor (J) is significant and reflects the shape of the tooth, the position at which the most damaging load is applied, stress concentration due to shape, and sharing of load. Reference is made to Figure 1 AGMA Standard for Surface Durability No. 220.02, August 1966.

J(pinion) = 0.360

J(gear) = 0.424

$$S_t\text{(pinion)} = \frac{300 \times 1}{1} \times \frac{12}{0.25} \times \frac{1 \times 1.5}{0.360} =$$

$$S_t = 60{,}000$$

COMPRESSIVE STRESS

The Compressive (Hertz) stress is calculated from the following formula:

$$S_c = C_p\sqrt{\frac{W_t C_o}{C_u}\frac{C_s}{D_f}\frac{C_m C_f}{I}}$$

where:

S_c = Compressive stress

C_p = Elastic coefficient

C_o = Overload factor = 1

C_u = Dynamic factor = 1

D_f = Pinion pitch diameter = 2.0 in.

C_m = Load distribution factor = 1

C_f = Surface condition = 1

I = Geometry factor = 1

C_s = Size factor = 1

When the factors C_o, C_u, C_s, C_m, C_f, and I are unity, the following formula can be used:

$$S_c = K\sqrt{\frac{W_t E}{2F}\left(\frac{r_2 + r_1}{r_2 r_1}\right)}$$

where:

K = Constant = 0.59

E = Modulus of elasticity = 30×10^6

r_2 = Roller radius of curvature of gear = 1.026 in.

r_1 = Roller radius of curvature of pinion = 0.342 in.

$$S_c = .59\sqrt{\frac{300 \times 30 \times 10^6}{2 \times 0.25}\left(\frac{1.026 + 0.342}{1.026 \times 0.342}\right)}$$

S_c = 157,000 psi

APPENDIX II

LITERATURE SURVEY

INTRODUCTION

The object of this literature survey was to find information related to solid lubricants that could be used in the development of an auxiliary solid lubrication system for helicopter transmissions. An auxiliary lubrication system is required that will permit continued operation of the helicopter transmission for a period of time after the conventional oil system is damaged during combat.

Vast quantities of information exist on the study of fluid lubricants and on initial studies of dry and solid lubricants, especially with respect to surface phenomena of wear and friction. Many of these studies were directed toward use in an aerospace environment.

Little information could be obtained on the development or use of solid lubricants in practical applications, either as the primary or the auxiliary lubrication system for operation at conventional temperatures. At present, three types of solid lubrication appear to be most promising: composite, powder, and thin film.

The composites (as noted in references 1 through 13) for both gears and bearings in general contain a matrix, a film former, and a load-carrying component. In applications where the loads are light or the speeds are low, one component can perform its function as well as that of one other. However, one component cannot replace the other two components unless other conditions are met. Further work in this field is still necessary to provide ample strength combined with adequate lubricating qualities. In cases where sufficient strength has been obtained, the lubricating properties have been diminished.

Different composites have been used for extremely heavy-loaded ball bearings, where loads have reached rated values that produced Hertz stresses of over 500,000 psi.[11] Heavy loads with tapered roller bearings were also attained.[23] In other applications, bearings have operated at high speeds for various temperatures.[12] In all of these tests, the retainers were made of a composite material.

Lubrication of gears appears to be more difficult, and less information is available. However, idlers of composite materials have been used to lubricate spur gears at low speeds and heavy loads where Hertz stresses were 200,000 psi.[11] A variation using composite lubrication is noted in tests where standard metal retainers were modified to use pockets filled with solid lubricants.

The second method of solid lubrication, that of the replenished powder technique, has been used for high-speed bearings[14] and spur gears[23] at elevated temperatures. Lubrication was accomplished by using an auxiliary injection system where air and powder were mixed and continuously metered into the bearings and gears.

The third type, thin film lubrication, has been used in several applications, although the lubricant is sacrificial. This method has shown considerable promise for small, lightly loaded ball bearings.[31] Such bearings are now used in aerospace applications and instrument bearings. Other work on thin film lubrication for heavy-loaded bearings and gears is also noted.[37]

The literature survey was expanded during its course to cover more than solid lubricants. As a result, many articles in other fields were screened to find information of pertinent value. The literature abstracts included in this report have been separated into four categories, each of which covers a significant aspect that can be useful in the present program:

1. Solid lubricants and lubrication systems.
2. Fluid lubricants and lubrication systems.
3. Gears.
4. Bearings.

SOLID LUBRICANTS AND LUBRICATION SYSTEMS

Composites

1. METAL-FILLED PLASTICS
   J. Delmonte
   Reinhold Publishing Corporation, New York, 1961, p. 240

   Discusses the preparation of finely divided metals and plastics, the processing of metal-filled plastic compositions; Al, steel, Cu, Pb, and Zn-filled plastics applications of metal fibers and filaments to plastics; bearing and friction materials; magnetic materials; electrical applications of plastics and metal alloys; and applications in nuclear reactors and radiation shields.

2. ANALYTICAL AND EXPERIMENTAL STUDY OF ADAPTING BEARINGS FOR USE IN AN ULTRAHIGH VACUUM ENVIRONMENT, PHASES I, II, AND III.
   P. H. Bowen
   ASTIA-AD 272555, 1963

   This report contains the results of an investigation into the lubrication of gears and bearings for use in a vacuum environment by using dry powder as a lubricant and dry self-lubricating materials in the bearing retainer. In Phase I, the wear and friction

characteristics of various dry powders and dryer self-lubricating materials for use in ball bearings were evaluated and screened in a dry, inert atmosphere in laboratory test apparatus under rotating speeds and loads similar to those found in 2 to 7 hp electric motors. The materials evaluated included reinforced thermosetting plastics, dry lubricant filled and unfilled thermoplastics, and dry lubricant filled sintered alloys. In Phase II, dry powder and self-lubricating materials were subjected to the vacuum conditions in the range of 1×10^{-6} to 1×10^{-9} mm Hg(torr) and at temperatures in the range of -60 to 1000°F to determine the rate of the outgassing and/or decomposition of each material. In Phase III, soaking and operating tests were conducted on dry ball bearings (204-size, 20 mm bore) using retainers fabricated from the most promising materials. The bearings were operated at a speed of 1800 rpm, a radial load of 75 lb, an axial load of 5 lb; they were tested under the vacuum and temperature conditions described in Phase II. Special bearings and retainer materials were used for exploratory tests of up to 1500°F.

3. DRY LUBRICATED BEARINGS FOR OPERATION IN A VACUUM
P. H. Bowen
ASLE Trans., 5(2):326, November 1962

Ball bearings incorporating two different types of dry self-lubricating retainer materials were successfully operated in a vacuum over the pressure range of 1×10^{-7} to 2×10^{-8} torr for prolonged periods of time. A 20 mm bore ball bearing of AISI M-10 tool steel with a filled plastic retainer was operated over two ranges of temperatures, from 100 to 160°F, and similar bearings with filled metal retainers were operated over a temperature range of -300 to 450°F, at radial loads up to 75 lb. Other ball bearings using both types of self-lubricating retainer materials were operated in electric motors in the vacuum environment.

4. MATERIALS SELECTION AND DEVELOPMENT FOR APPLICATION IN INTERPLANETARY VEHICLES
L. J. Bonis and G. S. Ansell
(Presented at the 9th National Vacuum Symposium--American Vacuum Society, 1962)

Investigations carried out in ultrahigh vacuum 1×10^{-10} to 1×10^{-11} torr range to select and develop materials applicable in a space environment are discussed. The materials tested, including alloys, intermetallics, and plastics and their use as bearings, gears, optical parts and materials are reported.

5. LONG TERM OPERATION AND PRACTICAL LIMITATIONS OF DRY, SELF-LUBRICATED BEARINGS
D. J. Boes
Lubrication Engineering, 19:137-142, April 1963

Three series of experiments which demonstrate the ability of a completely dry ball bearing to function satisfactorily for long periods of time under various combinations of load, speed, temperature, and atmospheric environment are described. The capability is achieved by equipping the standard ball bearing with a ball separator, or cage, that is fabricated from a material possessing inherent lubricating properties. The cage material used is reinforced polytetrafluoroethylene. The bearings operate from 1 x 10^{-5} torr to atmospheric pressure.

6. FRICTION-WEAR CHARACTERISTICS OF SELF-LUBRICATING COMPOSITES DEVELOPED FOR VACUUM SERVICE
D. J. Boes, P. H. Bowen
ASLE Trans., 6:192-200, April 1963

A research program designed to develop composite materials capable of functioning as a load-bearing surface with no lubrication other than that contained within itself is discussed. Described is a group of composites developed for ultrahigh-vacuum service, and their friction-wear characteristics in air and in vacuum are presented. The composites are shown to be capable of carrying loads of 1,400 psi with low self-wear and negligible wear of the sliding surfaces. Composed of polytetrafluoroethylene and a solid lubricant distributed throughout a metal matrix, the composites show good performance when tested as self-lubricating components in ball bearings, operating in environments that conventional lubricants cannot tolerate. The test environments include a vacuum of 1 x 10^{-8} torr and temperatures ranging from -180 to +400°F.

7. OPERATIONAL EVALUATION OF DRY-LUBRICANT COMPOSITES IN A HIGH VACUUM CHAMBER
A. G. Williams and T. L. Ridings
AEDC-TRD-63-67, May 1963
Arnold Engineering Development Center, Tennessee

This report contains the results of a test program to determine the operational characteristics of dry self-lubricating materials in the extremely low-pressure environment of a space simulator. The test was designed to evaluate the lubrication of gears, pinions, and bearings. Four selected self-lubricating composites fabricated as bearing retainers and idler gears were tested in the 7-ft aerospace research chamber at AEDC. Three of the composites consisted of a metal matrix, polytetrafluoroethylene (PTFE), and tungsten diselenide (WSe_2); the other consisted only

of PTFE and WSe_2. The three composites with the metal matrix performed satisfactorily; the fourth material did not provide an adequate lubricating film on the gears, which resulted in metal-to-metal contact and high wear.

8. SOME PRINCIPLES AND ADVANTAGES OF SUPER-REFLECTING SOLID MATERIALS FOR OUTER SPACE APPLICATIONS
L. J. Bonis and B. Manning
(Paper presented at NASA, DOD Space Lubrication Conference, May 3, 1963)

This report discusses the theory of solid lubricants and lubricating mechanisms in composite metal bonded solid lubricants. Microstructure and data on production and properties of these lubricants is also outlined.

9. AROMATIC POLYIMIDE COMPOSITIONS FOR SOLID LUBRICATION
M. J. Devine and A. E. Kroll
Lubrication Engineering, 20:225-230, June 1964

The mechanical and lubricant properties of an aromatic polyimide plastic are discussed. Data on such properties are given and compared with those of other materials. The discussed results of studies conducted at temperatures ranging from 77 to 700°F show that aromatic polyimide compositions are exceptionally heat stable, retaining good mechanical and electrical properties up to 700°F. Filled aromatic polyimide plastic is shown to be effective as a self-lubricating retainer in a ball bearing operating at temperatures up to 700°F and speeds of 10,000 rpm. The presented test results demonstrate that lubrication can be provided by the polyimide composition's functioning as the retainer. Bearing design considerations for solid lubrication extend the self-lubricating polymer to include usage as a land material. The importance of dimensions for components fabricated from the polyimide is noted.

10. DEVELOPMENT AND EVALUATION OF LUBRICANT COMPOSITE MATERIALS
M. E. Campbell and J. W. Van Wyk
Lubrication Engineering, 20:463-469, December 1964

Power metallurgy techniques were employed in the compounding and fabrication of lubricant composites, such as molybdenum disulfide in a metal matrix of iron and platinum, to obtain a suitable lubricant system for high-speed, high-temperature bearings operating in hard vacuum. Friction and wear tests have been conducted on these experimental composites at speeds of up to 7200 ft/min in air at both room temperature and 1500°F. Test results indicate

that this is a feasible approach to the dry lubrication of bearings. Fabrication procedures and friction and wear data for typical lubricant composites are presented.

11. SOLIDS AND SOLID LUBRICATION
M. J. Devine, E. R. Lamson, J. P. Cerini and R. J. McCartney
Lubrication Engineering, 21:16-26, January 1965

Presentation of a review covering solid-lubricant classification, methods of preparation, examples of engineering applications, and techniques for studying the different properties of solid lubricants, with extensive literature references. The results of new laboratory research are described, showing the degree of influence exerted by bearing design, lubricant-bearing interaction, and metal surface pretreatment on the wear properties of solid lubricants.

12. LUBRICATION OF BEARINGS AND GEARS FOR AEROSPACE FACILITIES
P. H. Bowen
AEDC-TR-65-91, April 1965
Arnold Air Force Station, Tennessee

This report describes the analytical experimental study of adapting lubrication techniques to satisfy the friction requirements associated with the vehicle handling system of the MARK I Aerospace Environmental Chamber. This report is in two parts. The first part covers adapting solid lubrication techniques for the rolling element bearings and wheel of the Horizontal Load Assembly and functionally evaluating the solid lubricated Horizontal Load Assembly in an ultrahigh vacuum under loads producing Hertz stresses in the bearings up to 500,000 psi. The second part of the report covers adapting solid lubrication techniques for the gears of the Roll and Pitch Assembly and functionally evaluating these solid lubricated gears in a four-square gear tester operated in an ultrahigh vacuum under loads producing Hertz stresses in the gear teeth up to 199,000 psi.

13. SOLID FILM LUBRICATION RESEARCH
D. J. Boes, E. S. Bober and K. W. Grossett
AFAPL-TR-66-110, Part I, October 1966
Wright-Patterson Air Force Base, Ohio

This report describes the first fifteen month's effort on a program that has as its objective the development of a self-contained, solid lubricated ball bearing system with long term operating capabilities at temperatures to 1500°F and speeds to 30,000 rpm. The bearing systems are to be capable of operating under these conditions in both an air atmosphere and an environment simulating a 200,000 ft altitude. The program concentrated

on optimizing the lubricating and mechanical properties of solid lubricant amalgams and on obtaining parametric design data relating ball bearing operating life with load, bearing size, bearing design, speed, temperature, and atmospheric environments. The maximum load encompassed by the program is a 100 lb thrust - 100 lb radial combination.

The work to date has resulted in the development of a 204 bearing system design capable of over 100 hours operation at 600°F and 10,600 rpm in an air environment. The bearings carried a load of 50 lb thrust - 50 lb radial. Lubrication was achieved by means of a self-lubricating retainer fabricated from a tungsten diselenide-gallium/indium composite.

14. UNIQUE SOLID LUBRICATING MATERIALS FOR HIGH TEMPERATURE-AIR APPLICATIONS
D. J. Boes
(Paper presented at ASLE-ASME Joint Conference in Minneapolis, Minnesota, October 18-21, 1966, Paper ASLE 66-LC-2)

A technique has been developed whereby mechanical strength and excellent oxidation resistance is imparted to self-lubricating bodies of high lubricant content. By an "amalgamation" of powdered solid lubricants, such as tungsten diselenide, with a gallium alloy, followed by a subsequent compression-curing cycle, self-lubricating surfaces have been found that resist oxidation at a temperature of 1500°F. The materials have demonstrated a load capacity of 1500 psi and good friction-wear characteristics over this range. They offer potential as load bearing surfaces and seals in high temperature applications for both oxidizing and inert or vacuum environments.

Powder Techniques

15. POWDER LUBRICATION OF ROLLING CONTACT BEARINGS AT VERY HIGH SPEEDS AND TEMPERATURE
A. L. Schlosser
(Presented at Proc. of USA Aerospace Fluids and Lubricants Conference, Southwest Research Inst., September 1963)

Investigations were conducted toward the immediate goal of attaining operation of an angular contact (20 mm bore) bearing over the temperature range of from room temperature to 1200°F at speeds of 50,000 rpm for periods to 10 hr. Evaluations of a four-to-one mixture (by weight) of graphite and cadmium oxide in an air environment indicate that this mixture is capable of lubricating titanium carbide bearings over the temperature range of from room temperature to 1200°F under combined loads of 50 lb radial and 50 lb thrust at speeds of 25,000 rpm.

16. AN INVESTIGATION OF THE LUBRICATION PROPERTIES OF MOLYBDENUM DISULPHIDE, PART I
E. Kay
Technical Note No. CHEM 1387 AD273474, November 1961
Royal Aircraft Establishment (Great Britian)

The frictional properties of MoS_2 were studied. The coefficient of friction of rubbed films of MoS_2 is dependent on humidity. When it was used as an additive to oils and greases, no benefit was obtained unless it adhered to the bearing surfaces. The lubricating properties were considerably modified by milling. Effective lubrication for sliding on Cu can be maintained with sintered compresses of mixtures of MoS_2, Ag, and Cu powders.

17. LUBRICATION STUDIES WITH LAMELLAR SOLIDS
P. J. Bryant
ADS TDR 62-55, January 1962
Wright-Patterson AF Base, Ohio

An investigation is being conducted to determine the mechanisms of friction and wear for lamellar solid lubricants. A basic research approach was applied to the problem. Molecularly smooth layers of mica were used for cohesion-adhesion experiments. Single crystals of graphite were produced. A series of experiments was conducted with molecularly smooth mica layers in vacuum and in air. The cohesion between clean degassed layers in ultra-high vacuum was measured. The same experiment with the same sample conducted in air yields a value of cohesion that is 30 times lower because of the action of atmospheric gas contaminants. From a series of air and vacuum tests, air exposure time, and separation rate studies, a stress-etch mechanism was proposed to explain the lower cohesion values observed in air. The mechanism obtained from mica studies may be applicable to other lamellar solids, such as graphite or molybdenum disulphide. This mechanism explains the effect of atmospheric molecules upon the lubrication properties of lamellar materials without requiring the molecules to penetrate the lattice or otherwise be present between the lamellae.

18. AN INVESTIGATION OF THE LUBRICATING PROPERTIES OF MOLYBDENUM DISULPHIDE, PART II
E. Kay
Technical Note No. CHEM 1397 AD 288615, June 1962
Royal Aircraft Establishment, Great Britian

An investigation is described of some chemical changes produced in MoS_2 by milling and their effect on the performance of MoS_2 as a lubricant. Results of a study of the large effect of relative humidity on the frictional properties of MoS_2 are also included.

19. ADAPTATION OF MoS_2 "IN SITU" PROCESS FOR LUBRICATING SPACECRAFT MECHANICAL COMPONENTS
Charles E. Vest
(Paper presented in AIAA 5th Ann. Structures and Matter Conference, 1964)

This "in situ" process consists of surface activation treatment - electrodeposition of an MoO_3 complex ion onto the substrate surface, and conversion of this film to MoS_2 in an atmosphere of H_2S gas at 400 psig pressure and 195°C, for an exposure period of 4 to 8 hours. From the work performed and the test results, it is concluded that: (1) the film thickness can be controlled within ± 35 micro-in.; (2) the average coefficient of friction of this film is 0.05 or less, and is comparable to or the same as MoS_2 powder, and is lower than bonded MoS_2 films; (3) the film can be easily and safely deposited onto a number of common spacecraft materials; (4) the film has a better wear life than burnished MoS_2 powder, and a somewhat poorer wear life than epoxy bonded MoS_2; and (5) the film follows the surface contour and fills up the smallest crack, lap, seam, or indentation.

20. PREVENTION OF CORROSION WHEN USING MOLYBDENUM DISULFIDE LUBRICANTS
Anonymous
Alpha Molykote Corp., Stamford, Connecticut, Spec. Print 477, 1964

Several factors were found that must be considered in the formulation of MoS_2-based, extreme-pressure greases and oils if the material is to have assured corrosion-preventive properties. First, the purity of the powder must be carefully controlled and then the factor of particle size must be evaluated. Finally, when required, the proper corrosion inhibitor or other additives must be selected and blended with the lubricant in the required proportions and compounded, not only so that the essential qualities of the lubricant are preserved but also so that the functional properties of the molybdenum disulfide are not impaired but, rather, are enhanced. Investigation of the merits of p-nonyl phenoxy acetic acid as a rust inhibitor was conducted. A low-concentration, fine-powder MoS_2 grease was selected as the subject of a corrosion test. One sample was blended with 1% of the inhibitor, and the other was used as a test control. The bearing lubricated with the inhibited grease showed no evidence of rusting, whereas the control bearing was partially corroded. It was, therefore, possible to nullify completely any rust-inducing tendencies of MoS_2 greases.

21. HIGH-TEMPERATURE LUBRICATION AND INORGANIC SOLID SUBSTANCES
Ch'en Shao-li and Ou-yang Chin-lin
Joint Publications Research Service, Washington, D.C.
(JPRS-23893; OTS-64-21906)

A brief report is presented of investigations of the friction-temperature characteristics of various types of inorganic solid powders. The substances included high-melting-point metal oxides, sulfides, salts (inorganic and organic), silicates, graphite, and carbon black. The test results indicate that the selection of multimixture inorganic solid substances is one hopeful means of maintaining low friction and high resistance against wear for high-temperature solid lubricants of wide temperature ranges.

22. THE EFFECTS OF LOAD ON THE FRICTIONAL PROPERTIES OF MOLYBDENUM DISULFIDE
S. A. Karpe
(Presented at ASLE-ASME Lubrication Conference, Washington, D.C., October 13-16, 1964, Paper ASLE-64-LC-2)

A brief summary is made on the work of M. J. Devine of US Naval Air Engineering Center; Josef Gonsheimer of West Germany and G. Saloman and A. W. J. de Gee of Delft, Netherlands, regarding the interrelation of load on the frictional properties of MoS_2 powder.

23. FUNDAMENTAL INVESTIGATION OF MOLYBDENUM DISULFIDE AS A SOLID LUBRICANT
J. C. Tyler and P. M. Ku
Final Report Contract NO64-0545-C, RS-460, August 1965
Bureau of Naval Weapons, Washington, D.C.

This report describes the third three months of work on an investigation of molybdenum disulfide (MoS_2) as a solid lubricant. The objective or documented composition and particle size distribution has been used for making a limited number of preliminary compression tests. The relation between specimen fracture stress, specimen specific gravity, and specimen length-to-diameter ratio has been investigated and is discussed. The variations of specimen specific gravity and hardness in the longitudinal direction have also been studied and the results are presented. A crack developed in the four-piece tapered sleeve of the compacting die during compact pressurization, and a new sleeve is in the fabrication stage. Fabrication and assembly of the test apparatus have been completed; the test apparatus is being instrumented and calibrated for subsequent work.

24. **DEVELOPMENT OF GAS ENTRAINED POWDER LUBRICANTS FOR HIGH SPEED AND HIGH TEMPERATURE OPERATION OF SPUR GEARS**
A. I. Schlosser
DDC Control No. 030770, AF 33(657)8625
Defense Documention Center, Cameron Station, Alexandria, Va.

Spur gears were successfully operated at a constant tooth temperature of 89°F for 24 hr. Another successful evaluation of approximately 45 hr duration was cycled through 16-1/2 temperature cycles from ambient temperature to a tooth temperature of 900°F. Loading was 1000 ppi (defined as pounds per linear inch of tooth face), and speeds were 7400 rpm (6100 fpm pitch-line velocity). The gear material was M-50 tool steel that was heat-treated to a minimum hardness of R_c 60. The lubricant was a powder composed of five parts of micronized Acheson No. 38 graphite pulse to one part of technical laboratory grade cadmium oxide entrained in an air carrier.

25. **DEVELOPMENT OF GAS ENTRAINED POWDER LUBRICANTS FOR HIGH SPEED AND HIGH TEMPERATURE OPERATION OF SPUR GEARS**
S. Wallerstein
AFAPL-TR-65-24, May 1965
Wright-Patterson Air Force Base, Ohio

The feasibility of adapting powder lubricants to the operation of gears during relatively long periods of time under extreme environmental conditions was established. In addition to the lubricant study, parallel investigations were conducted on gear materials and methods of dispensing powder lubricants. A pair of 5 diametral pitch (DP) spur gears, manufactured from M-50 tool steel, had operated for 98-1/2 hr at a speed of 7400 rpm, with a load of 1000 lb per linear inch of tooth face, and with the temperature cycled from room temperature to 900°F. Evaluations of fine-pitch (12/14 DP) superalloy and toolalloy steel gears were conducted at speeds of up to 15,500 rpm, at temperatures in excess of 1000°F, and with loads of up to 1000 lb per linear inch of tooth face. All high-temperature evaluations performed during this program used a graphite plus cadmium oxide powder mixture as the gear lubricant. An air carrier was used to deliver the powder to the gear set.

26. **FUNDAMENTAL STUDIES OF COMPRESSIBILITY OF POWDERS**
I. I. Shapiro
DDC Control No. 030770
Defense Documentation Center, Cameron Station, Alexandria, Va.

The compressibility of a number of powders of different physical characteristics was measured at ambient temperature. The progress of the compaction behavior of the powders was followed by photomicroscopy; this technique permitted observation of changes

occurring during compression. The difference between plastic deformation of metals, such as magnesium, and the fragmentation of ceramics, such as thoria, could be clearly demonstrated. The compressibility of molybdenum disulfide, a material commonly used as a lubricant, indicated a high degree of "plastic" quality as well as some fragmentation. Mixtures of molybdenum disulfide with ceramics resulted in both types of compaction. The applicability of several formulas relating porosity or density of compacts with pressure is discussed. The results of the present study are regarded as most significant toward clarifying erroneous hypotheses as to what forces are operating during compaction of powders. The concept of particles sliding past one another is contrary to experimental findings. The concept of mechanical interlocking of particles as an explanation of strength of compacts does not appear to be valid.

Thin Film Lubrication

27. INORGANIC SOLID FILM LUBRICANTS
M. J. Levine, E. R. Lamson, and J. H. Brown
Journal of Chemical and Engineering Data, 6:79-82, 1961

An investigation of inorganic solid film lubricants, derived from $Na_2O:SiO_2$-type binders and solids, such as MoS_2, graphite, AgI, etc., under extreme environmental conditions is discussed. A means was derived to utilize these types of lubricants in high-speed ball bearing applications.

28. POLYMER FORMATION ON SLIDING METALS IN AIR SATURATED WITH ORGANIC VAPORS
W. E. Campbell and R. E. Lee
ASLE Trans., 5:91-104, April 1962

A study is made of lubricating properties of polymer formed in wear track by mechanical activation during continuous slide between Pd and Pd, Cr and Cr, and steel and steel. Contrary to expectations, polymer, which forms on Pd in air saturated with limonene and diisobutylene, does not appear to reduce wear appreciably; the nature of absorbed film, which precedes formation of polymer, is the main determining factor in wear and friction reduction.

29. GALLIUM-RICH FILMS AS BOUNDARY LUBRICANTS IN AIR AND IN VACUUM TO 10^{-9} mm Hg
D. H. Buckley and R. L. Johnson
ASLE Trans., 6:1-11, January 1963

The friction and wear characteristics of various materials coated with thin gallium-rich films were determined at temperatures of up to 1000°F in air and at room temperature in a vacuum between

10^{-7} and 10^{-9} torr. Evaporation rates of gallium were measured at 10^{-7} torr and ambient temperatures of up to 1000°F. The friction and wear experiments were conducted with a 0.187-in. radius hemispherical rider sliding on a 2.5-in. disk at surface speeds of 28 to 4490 ft/min and a load of 1000 gm. Boundary lubrication of 440-C stainless steel was obtained using a gallium-diffused film. Gallium was not equally effective as a lubricant for all materials; it reduced friction and wear for several alloys (52100 and 440-C), but other materials, including a nickel-base alloy, were not effectively lubricated.

30. SOLID FILM LUBRICANT-P[illegible]ER PHENOMENA: $PbS-B_2O_3$ SYSTEM
H. R. Thornton, Doris M. Kumwiede, et al.
ASD TDR 62-449, AD-278822, May 1962
Wright-Patterson Air Force Base, Ohio

Basic techniques, X-ray diffraction, microscopy, and fucion studies, with the supplementary techniques of differential thermal analyses and friction and wear measurements, are described as related to the binary phase system, $PbS-B_2O_3$. The conversion of PbS to $PbS-B_2O_3$ mixtures indicated that PbS and B_2O_3 were the only crystalline phases to be expected in the majority of the specimens. A glassy phase exists in all specimens, to some degree, over the temperature range to 1500°F. The mechanism of lubrication in the high B_2O_3-low PbS mixtures is a function of the amount of liquidus phase present while this liquidus phase affects only the low B_2O_3-high PbS mixtures above 980°F. For a good operational solid-film lubricating system, frictional compatibility must be observed between the lubricating pigment and the binder over the entire temperature range.

31. THIN FILM LUBRICATION
V. H. Brix
Lubrication Engineering, 18:312-319, July 1962

Analysis of the behavior of a wide range of bearing materials under thin film conditions reveals that certain metal pairs, when run together, evidence an inherent predisposition to favor hydrodynamic oil films between them.

32. SOLID FILM LUBRICATED BEARING RESEARCH PROGRAM
Paul Brown, Roger M. Hawkins, M. Maguire and M. Pitek
FDL-TDR-64-117: AD-608629
Wright-Patterson Air Force Base, Ohio

The program was divided into two phases. The first phase consisted of analytical evaluations of potential bearing designs, solid lubricant materials and lubricant supply systems as applied to accessory-sized ball bearings. The second phase was devoted to laboratory determinations of lubricant materials tensile strength

and thermal properties followed by evaluations of selected solid lubricant materials and bearings. These tests included friction and wear evaluations of both lubricant and bearing materials in a friction and wear rig; evaluations of full-scale bearings in atmospheric and vacuum environments at temperatures ranging from -249 to 1500°F, at speeds up to 24,000 rpm for periods up to 3 hr, and vacuum-environment testing of bearings that were exposed to a fast neutron dose of approximately 1.0 x 10^{17} neutrons/cm^2 (> 1.0 MeV). Test results indicated that no lubricant tested, regardless of the supply system used or conditions under which tested, showed any clear superiority when compared to other dry film lubricants.

33. DRY FILM LUBRICATION OF HIGHLY LOADED BEARINGS IN VACUUM
E. C. McKannan and K. E. Demorest
Lubrication Engineering, 20:134-141, April 1964

Results are presented of tests of 13 dry film lubricants in a gimbal simulation device. The problem was posed by the necessity of lubricating gimbal bearings for the upper stages of boost vehicles operating in vacuum and under high loading. The lubricant was required to be nonvolatile and resistant to temperature variations, vibration, and nuclear radiation. It has to perform satisfactorily under slow oscillating motion, to resist cold welding during quiescent periods, and to permit restarting. The test apparatus duplicated flight conditions for bearing contact load, bearing materials, type of motion, and environment. Of the lubricants tested, the following two provided the necessary properties: (1) a mixture of molybdenum disulphide, graphite, and gold powders with a binder of sodium silicate, capable of being sprayed with an air gun and (2) a flame-sprayable coating of zirconium silicate as a binder for burnished molybdenum disulphide powder.

34. A REVIEW OF TECHNIQUES FOR INVESTIGATION OF FRICTION AND WEAR IN AEROSPACE BALL AND ROLLER BEARINGS
L. C. Lipp and E. N. Klemgard
American Society of Lubrication Engineers
Lubrication Engineering, 19:495-502, December 1963

Review of analytical and experimental methods for isolation and investigation of 21 parameters of importance in friction and wear of aerospace ball and roller bearings when operating under conditions of -65 to +1500°F, 10^{-9} mm Hg pressure, and variable loads and speeds. It is shown that the use of infrared electron microscopy, high-speed photography, radioactive isotopes, radiation pyrometers, and friction measuring oscilloscopes are instrumental in the study of these friction and wear parameters. Application of a thin adherent lubricant coating to the ball and roller bearing surfaces is discussed, and six possible methods by which this may be accomplished are covered.

35. AIR FORCE MATERIALS LABORATORY SOLID FILM LUBRICATION RESEARCH
B. D. McConnell
ML-TDR-64-46: AD-604654
Wright-Patterson Air Force Base, Ohio

This report outlines the general range of bearing lubrication requirements of present aerospace vehicles and briefly describes the range of requirements anticipated for future Air Force weapon systems. This report covers only those programs concerned with solid film lubrication and contains discussion of both development-type programs and related basic or fundamental-type programs. Program objectives, approaches to the problems, data obtained to date, and future work are discussed for each program.

36. RESEARCH ON BEARING LUBRICANTS FOR USE IN HIGH VACUUM
Vern Hopkins and D. H. Gaddis
NASA-CR-58204: OTS: August 1963

Binder development studies were performed to develop better binders for solid lubricant films during the past year. Potassium silicate was selected as a basic binder. Preliminary results show that the wear life of this binder material may be increased by additives such as sodium phosphate, potassium phosphate, sodium borate, or sodium fluoride. A search for additional lubricants or lubricant film components was conducted. Nine potential materials were selected for formulation and evaluation. A gear apparatus and pellet apparatus were designed, built, and used to investigate solid lubricant film wear life in air at room temperature. Wear characteristics of MLF-5 (MoS_2 + graphite + gold-sodium silicate) obtained in the early runs with the pellet apparatus are presented graphically. MLF-5 and other solid lubricant films were applied to a number of parts and components. A number of modifications that were made on the vacuum friction apparatus to improve its overall operating efficiency are described.

37. FACTORS INFLUENCING FRICTION OF PHOSPHATE COATINGS
Michael A. George
SA-TR18-1084: AD-603320, April 1964
Springfield Armory, Massachusetts

Coefficients of friction for the coatings were determined under static and dynamic conditions. The following factors influencing the coefficient of friction are considered: type of coating, lubrication, loading weight, surface roughness, crystalline structure, and velocity. The coefficients of friction in manganese phosphate coatings did not differ to any practical extent from the coefficients for zinc phosphate coatings. Lubrication is a significant factor on the coefficients of friction for phosphate coatings. The coefficient of friction was independent of the applied load. Velocity during dynamic testing, surface finish, and crystalline structure influenced the coefficient to a slight degree.

38. LUBRICATION OF HEAVILY LOADED, LOW VELOCITY BEARINGS AND GEARS OPERATING IN AEROSPACE ENVIRONMENTAL FACILITIES
R. E. Lee
AEDC-TR-65-19, January 1965
Arnold Air Force Station, Tennessee

Both bearings and gears were lubricated with thin films. Molybdenum disulfide + graphite + sodium silicate binder functioned well. Operating speeds were from 8 to 11 rpm in a vacuum at pressures in range of 10^{-7} to 10^{-8} torr. Bearing load ratios were 10 to 18 (static radial load to applied radial load capacity); loads on the gears were such that the unit compressive stresses in the teeth were 47,000 to 67,700 psi.

FLUID LUBRICANTS AND LUBRICATION SYSTEMS

39. EFFECT OF SPACE ENVIRONMENT ON THE EXTREME PRESSURE QUALITIES OF LUBRICANTS
J. G. Williamson and E. E. Nelson
Final Report, Part I, ABMA-5, October 1960
Army Ballistic Missile Agency, Huntsville, Alabama

The object of this work was to study the lubricating qualities, both extreme pressure and wear characteristics, of mineral and synthetic oils with various general types of extreme pressure additives in inert gases and at reduced pressures.

40. A STUDY OF THE INFLUENCE OF LUBRICANTS ON HIGH-SPEED ROLLING-CONTACT PERFORMANCE
Lewis B. Sibley, J. Clarence Bell, et al.
ASD TR 61-643, pt. 2 AD-292666, November 1961
Wright-Patterson Air Force Base, Ohio

The thickness of the lubricant film and the shape of the elastically deformed surfaces at rolling contacts, as measured by an X-ray method, indicate that some non-Newtonian flow properties of lubricants may have important effects in rolling-contact lubrication. A high-pressure lubricant rheology device is being developed to measure these properties under simulated rolling-contact conditions. Several difficult equipment and instrumentation problems were solved to obtain accurate shear-stress and shear-rate data on the rheology machine. However, a brief consideration of some remaining thermal problems in the rheology machine has led to the temporary abandonment of this experimental approach to obtaining realistic high-pressure rheology data on lubricants in favor of a proposed new approach to the analysis of the disk-machine data. The disk machine has been instrumented to obtain the traction or friction at the rolling contact over a range of known amounts of slip superimposed

on the rolling. Initial traction data, together with an analysis for interpreting these data in terms of pressure-viscosity coefficients, indicate considerable promise for the rheological technique.

41. INFLUENCE OF LUBRICATION ON ENDURANCE OF ROLLING CONTACTS
T. Tallian, Y. P. Chiu and E. F. Brady
AL63T018: AD-417518, 1963
Bureau of Naval Weapons, Washington, D.C.

Conductivity tests were conducted at low and high speeds in order to determine speeds, loads, and viscosity parameters for conditions of no film and essentially complete film lubrication, respectively, at the ball-to-ball contacts in a rolling four-ball test configuration. Data from these tests are still in the process of interpretation. The dynamic two-ball tester, designed to provide X-ray measurements of lubricant film thickness and contact profiles in a two-ball rolling contact, is described. Details of the X-ray mechanism, drive system spindles, fabrication and alignment procedure, the hydraulic system, and the electrical control system are given.

42. BEARING LUBRICATION UNDER SEVERE CONDITIONS
J. B. Accinelli, W. M. Widlund, and W. W. Kerlin
S-13918: AD-440305, May 1964
Bureau of Naval Weapons, Washington, D.C.

The studies include a realistic evaluation of several lubricants in a 35-mm ball bearing rig operating at severe conditions of 40,000 rpm speed, 400-lb thrust load, and 400 to 800°F temperature. The lubricants tested were a highly refined mineral oil, a MIL-L-7808E lubricant, a MIL-L-9236B lubricant, and a polyphenyl ether (mixed 5P4E) lubricant. The best performance was obtained with the 5P4E at 600°F (66 hr), whereas at 800°F, operation with this lubricant became critical. For a system that fails by loss of lubricant, two equations were derived that relate initial lubricant charge, termination lubricant charge, lubricant recovery rate, and lubricant flow rate with total operating time and number of lubricant passes. The experimental values of time and number of passes for the current tests are in excellent agreement with values predicted by the derived expressions.

43. ELASTOHYDRODYNAMIC LUBRICATION-EXPERIMENTAL INVESTIGATION
F. K. Orcutt
MTI-64TR6: AD-432340, February, 1964
Office of Naval Research, Washington, D.C.

The overall objective of this program is to improve the life, reliability, and load capacity of concentrated contact machine elements, such as rolling-element bearings, gears, and cams. The chosen means of accomplishing this objective are to investigate the fundamental process of lubrication in concentrated

rolling-sliding contacts and to attempt, wherever possible, to relate the findings to the practical aspects of bearing, gear, and cam performance. Measurements of several of the important variables of the load zone of the rolling-sliding disk apparatus were made. Measurements of the temperature of the surface as it moves through the load region were obtained for light-to-moderate loads with slip ratios up to about 25 percent, using improved vapor-deposited thermocouples and the rolling-disk apparatus.

44. LUBRICANTS AND LIQUIDS FOR MOTORS AND JET ENGINES (MOTORNYE I REAKTIVNYE MASLA I ZHIDKOSTI). 4th EDITION
K. K. Papok and E. G. Semenido
Moscow, Izdatel'stvo Khimiia, 1964

This book discusses the fundamentals of friction and lubrication, and reviews methods of evaluating the physical and chemical properties proper--as functions of hydrocarbon structure. Experience obtained in the production of hydrocarbon lubricants and of synthetic lubricants without hydrocarbon content is noted. Particular attention is given throughout to the effects of various additives. The stability, fractional composition, and thermal properties of lubricants are examined, as are the mechanisms of formation of carbon-black and other sediments in engines, including the method of removing the latter. Several chapters are devoted to the quality and selection of lubricants and liquids for use in aircraft piston and jet engines, aircraft and conventional gas turbines, as well as engines of automobiles and ships. Some aspects of the recovery of used lubricants are noted. The book is designed for engineers and technicians working in the lubrication field.

45. DETERMINATION OF THE AXIAL CONSUMPTION OF THE LIQUID DURING THE ROTATION OF A SHAFT (OPREDELENIE OSEVOGO RASKHODA ZHIDKOSTI PRI VRASHCHENII VALA)
A. I. Belousov
Aviatsionnaia Teknika, 7:3, 1964

This report sets forth a determination of the consumption of a lubricant along the bearing's axis in a turbulent flow, considering the shaft's rotation. Expressions for lubricant losses in the cylindrical annular groove of hydrostatic bearings are derived on the assumptions that (1) the flow rate in the groove is stable; (2) the flow rate in a cavity perpendicular to the flow motion is constant and equal to a flow rate at which there is average lubricant consumption; (3) the groove is entirely filled with the liquid; (4) the coefficient of friction is independent of the rotation rate; and (5) rotation does not affect the degree of low turbulence.

46. ADVANCED LUBRICANTS AND LUBRICATION TECHNIQUES
R. Adamczak, R. Benzing and H. Schwenker
Industrial and Engineering Chemistry, 56, January 1964

Lubricants and lubrication techniques able to meet the requirements of the space age are discussed. Considered are specially processed mineral oils and synthetic hydrocarbons, esters, silicon-containing fluids, polynuclear aromatics, advanced fluids, liquid metals and salts as lubricants, gas lubricated bearings, solid films, dry lubrication, in situ films, and miscellaneous techniques. It is shown that there is a wide variety of lubrication techniques for potential use at elevated temperatures. It then becomes a major problem to select the correct system for use in advanced vehicle. It is noted that it is an even greater problem to select those systems which offer promise for future use and which should receive the major research effort.

47. LUBRICATION REQUIREMENTS FOR SPACE ENVIRONMENTS
D. J. Pinson and W. F. McRae
DDC Control No. 030770
Defense Documentation Center, Cameron Station, Alexandria, Va.

The present space age has brought to light a new era in the field of lubrication for today's aerospace systems and facilities. Development of special lubrication techniques has become essential to the successful operation of these advanced systems and facilities. In the normal design or manufacture of machines, lubrication is a secondary consideration. However, for the operation of moving mechanisms in the space environment, lubrication must be considered concurrent with the design of the system. This paper presents a concise review of present work being undertaken by leading scientists throughout the country in development of space lubrication. It presents a detailed description of research undertaken by the Arnold Engineering Development Center in this area, with special emphasis on results of tests related to operating gears and bearings in a simulated space environment.

48. PREPARATION OF TABLES OF COMPARISON FOR AMERICAN, BRITISH, AND CANADIAN OILS, HYDRAULIC FLUIDS AND SPECIALTY PRODUCTS
E. Ewell
DDC Control No. 030770
Defense Documentation Center, Cameron Station, Alexandria, Va.

The compilation was limited to oils, hydraulic fluids and specialty products. Specifications were obtained for the applicable American, British and Canadian products listed in the "Inter Changeability Chart of NATO Standardized Air Fuels, Lubricants and Allied Products". A table was prepared for each standardized lubricant, comparing the individual specifications. For the purposes of the Air Standardization Coordinating Committee, products of two or more countries

are considered to be standardized when they conform to national specifications having the same technical requirements. Such standardized products are allocated the same NATO symbol. Minor differences between standarized specifications are noted.

49. LUBRICATION RESEARCH AND TEST METHOD DEVELOPMENT FOR AEROSPACE PROPULSION SYSTEMS
B. B. Baber and F. Chang
DDC Control No. 030770
Defense Documentation Center, Cameron Station, Alexandria, Va.

The work reported is concerned with lubrication problems associated with advanced primary and secondary propulsion systems for aviation and space applications. The program was divided into five principal phases of investigation: gear lubrication, bearing lubrication, lubricant bench tests, impact sensitivity of materials in contact with strong oxidizers, and a lubricant compatibility survey. The effort was devoted largely to the development of suitable organic liquid lubricants intended for use in high-temperature applications, to the modification of an ABMA tester for evaluating the impact sensitivity, and to the completion of a special report dealing with the test methods for determining the compatibility of lubricant materials with rocket propellants.

50. AN INVESTIGATION OF OIL FILM THICKNESS BETWEEN INVOLUTE GEAR TEETH
I. D. Macconochie and L. C. Hsu
DDC Control No. 030770
Defense Documentation Center, Cameron Station, Alexandria, Va.

Investigation of oil film thickness between involute gear teeth was determined, using electric potentials. Fluid dynamic properties as well as polyamides were also studied.

GEARS

51. STUDY OF HELICOPTER GEAR LUBRICATION
S. J. Beaubien, L. Lichtman, C. A. Converse, et al
Contract NOw63-0557-C, May 1964
Bureau of Naval Weapons, Washington D.C.

The study on helicopter gear lubrication concerns the types of failure that have occurred in helicopter transmissions in the field; and an attempt was made to learn the causes of such failures on a theoretical and experimental basis. The overall objectives of the work, however, are not limited to solutions of gear lubrication problems of existing helicopter transmission designs; in part, the solutions are applicable to other gear systems.

52. MATERIALS AND RATINGS FOR DRY-RUNNING GEARS
H. J. Watson
David Brown Industries Ltd., England, December 1961

The requirements of nuclear power stations have led to the development of gears intended to transmit measurable power when running dry. For some applications, a restriction on the use of a range of metals promoted an investigation into non-metallic materials. Such requirements tended to encourage the design of gears possessing a low power rating owing to difficulties connected with adequate heat dissipation. Experimental work necessary to determine a satisfactory material combination for worm gears operating without lubrication is described for discs and gears. Further experimental results on work undertaken to meet industrial requirements for spur gears running with a dry lubricant have been reported, and the limitations of the materials used have been shown. The effect of different surface treatments applied to gear-tooth flanks has been investigated, and findings are described. A comparison made between the Novikov and involute tooth forms on double helical gears running dry has shown the latter to be superior for wear resistance under the test conditions used. Practical methods of establishing permissible rating criteria are described.

53. EXPERIMENTAL INVESTIGATION OF THE MINIMUM/OIL-FILM THICKNESS IN SPUR GEARS
D. W. Dareing and E. I. Radzimovsky
ASME Transactions, Series D, 85: 451-455, September 1963

A pair of gears is loaded; the minimum oil-film thickness between the gear teeth decreases and can approach a magnitude equal to the magnitude of the surface roughness. Metal-to-metal contact then occurs between the microscopic peaks on both mating teeth surfaces. Therefore, the minimum thickness of the film separating the mating teeth surfaces may be considered as one of the criteria of capacity for a gear drive. A testing technique that was developed for measuring oil-film thickness between loaded gear teeth while running is presented. The voltage drop across a thin oil-film that is required to cause an electrical discharge is used to determine the oil-film thickness. A specially designed machine containing a planetary gear train is employed in these experiments. The relationships between the minimum oil-film thickness and the load transmitted by the gearing under certain conditions are determined using this method.

54. WEAR ANALYSIS OF NONLUBRICATED SPUR GEARS
J. C. Randall
NASA CR-53197: JPL-TM-33-139, June 1963
NASA, Goddard Space Flight Center, Greenbelt, Maryland

This paper established a method of determining wear rates for nonlubricated, fine-pitch, precision instrument spur gears. The concepts of wear and the problems associated with applying these concepts to the unique action of spur-gear surfaces are discussed. Wear data for test gears run at various loads and speeds are collected to determine the wear rates for the most popular materials in use today. A method is proposed for using the wear data to select between two popular methods of computing dynamic load; namely, the American Standards Association Specification B6ll-1951 and Tuplin's method, both of which are slight modifications of Buckingham's original spur-gear formulas.

55. TESTS OF DRY COMPOSITE LUBRICATED GEARS FOR USE IN AN AEROSPACE ENVIRONMENTAL CHAMBER
T. L. Ridings
AEDC-TR-65-45: AD-460502, March 1965
Arnold Air Force Station, Tennessee

The results are reported of a test program set up to determine the operational characteristics of dry composite lubricated gears. Two diametral pitch sizes, 7 and 12, and two gear materials, nitralloy steel and nodular iron, were tested. Three dry composite lubricants and one low vapor pressure grease were tested. All three dry composite lubricants provided adequate lubrication for periods of up to 300 hr at 100 rpm with very little wear of either load gears or lubricating idlers The MoS_2-fortified, grease-lubricated gears failed after 40 hr of operation.

56. OGO WABBLE GEAR MATERIAL AND LUBRICATION EVALUATION TESTS
A. Ambruso and J. Heindl
DDC Control No. 030770
Defense Documentation Center, Cameron Station, Alexandria, Va.

The report discusses an evaluation of gear material and lubricants under vacuum conditions. Special coatings of gold, molybdenum compound, and other plating materials along with gear materials of steel, bronzes, aluminum alloys, and powder alloys were used.

57. INVESTIGATION OF VARIOUS LUBRICANTS FOR USE IN GEARBOXES
R. Smith, Eastman Kodak Company
DDC Control No. 030770
Defense Documentation Center. Cameron Station, Alexandria, Va.

This report describes an investigation of Teflon coatings, oil, and grease as lubricants for a 3-stage planetary gear system with a 121-to-1 reduction.

BEARINGS

58. INVESTIGATION OF COMPLEX BEARING AND/OR LUBRICATION SYSTEMS FOR HIGH SPEED, HIGH TEMPERATURE OPERATION
P. Lewis, S. F. Murray, and M. B. Peterson
FDL-TDR-64-12: AD-273864, January 1964
Wright-Patterson Air Force Base, Ohio

This report describes a program whose objective was to use complex or combined systems to permit operation over a wide temperature range at high speed and with a variety of ambient pressures. The program reviewed in detail the various individual items that constitute an overall system. Based upon the requirements and the results of the review, a rolling element bearing system with a solid lubricant circulating system was selected. Experimental results are presented that explore the feasibility of the elements making up the overall system.

59. TEMPERATURE LIMITATION OF PETROLEUM, SYNTHETIC AND OTHER LUBRICANTS IN ROLLING CONTACT BEARINGS
Z. Nementh and W. Anderson
SAE Transactions, 63: 556-565, 1955

Studies of 20 mm bore, tool-steel ball bearings operated at 2500 rpm and lubricated with graphite, molybdenum disulfide air mist, or one of several liquids are reported. Effective lubrication at 1000°F with graphite and at 700°F with molybdenum disulfide and synthetic diester was obtained. A second investigation of the effectiveness of molybdenum disulfide air mist as a lubricant for high-speed roller bearings is reported.

60. DESIGN FOR ROLLING ELEMENT AIRFRAME BEARINGS FOR HIGH TEMPERATURE AND HIGH ALTITUDE USE
J. B. Havewala and J. H. Johnson
Lubrication Engineering, 20: 25-33, January 1964

This paper discusses an experimental investigation for the evaluation of bearing materials and lubricants for operation in the temperature range of 1200°F at a simulated altitude of 250,000 ft. Four different roll designs, together with 12 different material

combinations, were investigated. The resulting best design and the two best material combinations were subjected to stresses of up to 325,000 psi average Hertz stresses at the temperature and vacuum. One-inch-diameter bore, self-aligning, double-row roller bearings fabricated from Wrought alloy carried loads of up to 5,000 lb (280,000 psi average Hertz stresses) for 40,000 cycles at 1200°F and vacuum. All contact surfaces were treated with DF-700 dry film (MoS_2 + graphite + sodium silicate). Friction coefficients with dry lubricants were in the range of 0.08 to 0.25. It is noted that successful bearing operation required considerable deviation from design criteria for fluid lubricated bearings.

61. BEARINGS FOR VACUUM OPERATION RETAINER MATERIAL AND DESIGN
H. E. Evans and T. W. Flately
TND-1339, May 1962
NASA, Goddard Space Flight Center, Greenbelt, Maryland

Fully machined retainers of five different materials, with all balls and races of gold-plated 440C stainless steel, were tested. Both pure gold plating and gold with additives were investigated. Size R2-5 bearings were run without external loading at a nominal motor speed of 10,000 rpm, and the goal is a bearing life of 1,000 hours in an ambient pressure of 10^{-7} torr. The results show that: (1) thin metallic films as lubricants show real promise when used in a vacuum environment; (2) pure gold plating is not as effective as the plating with additives; (3) fully machined retainers provide good performance, and the use of relatively hard retainer materials significantly extend the useful life of the bearings; and (4) the bearing failures tended to be catastrophic rather than gradual, making the prediction of the onset of failure difficult. A special multiport oil-free vacuum system designed and built for this program proved extremely effective in achieving a vacuum of 10^{-7} torr and in permitting the operation of seven individual tests at one time.

62. DEVELOPMENT OF DESIGN CRITERIA FOR A DRY FILM LUBRICATED BEARING SYSTEM
M. E. Campbell and J. W. Van Wyk
ASD-TDR-62-1057, March 1963
Wright-Patterson Air Force Base, Ohio

This research was initiated to determine the extent to which dry lubricant films could be used in future bearing systems for electrical accessory applications.

In Phase I, dry film lubricated plain, ball, and roller bearings were tested in 900°F air at 15,000 rpm. Two different bearing designs, which used unconventional dry film lubrication techniques, demonstrated the feasibility of operation at 15,000 rpm in 900°F air.

In Phase II, roller and ball bearings were evaluated through the temperature range of 70 to 1500°F at 15,000 rpm in a vacuum. An investigation was initiated to develop new lubricant composite materials for dry film lubrication under vacuum conditions. Conception of a new and unique bearing design, utilizing a lubricant composite material as the cage, resulted in successful vacuum operation for both ball and roller bearings.

63. SOLID FILM LUBRICATED BEARING RESEARCH
P. Brown, R. Hawkins, M. Maguire, et al.
FDL-TDR-64-117, October 1964
Wright-Patterson Air Force Base, Ohio

This report covers the analytical and experimental evaluation of solid-film lubricated accessory-size ball bearing designs under simulated space conditions. The desired operating conditions included speed levels of 0-24,000 rpm, temperatures of -200°F to 1500°F and environmental pressures of 760 mm Hg to 10^{-9} mm Hg. The Contract also required that the effects of nuclear radiation on the chosen substrate and solid film lubricant materials be determined. The aspects of design considered were: (1) the ball, race and cage substrate material; (2) solid film lubricant type and method of application; (3) internal geometry; and (4) means of automatically supplying and replenishing the solid lubricant material.

Generally, for an application of this type where the bearing is required to supply its own lubrication, the bearing retainer is the critical element. This was borne out by preliminary vacuum bearing testing at room temperature and moderate speeds of up to 8000 rpm on a 208-size counterbore ball thrust bearing run in a company-sponsored program. Work on the development of a cryogenic bearing of this same type and size for operation at 30,000 to 40,000 rpm also served to emphasize the critical need for a strong, self lubricating retainer design. As a result of this experience and that of other investigators, who found that the most success was achieved for ball bearing operation in vacuum by means of self lubricating retainer designs, a great deal of emphasis was placed on this approach.

64. BALL BEARING PERFORMANCE IN HELIUM ATMOSPHERE AT 500°F
Jamshed B. Havewala and John J. Johnson
(Presented at the ASME Winter Annual Meeting held at
New York, New York, November 29-December 4, 1964
Paper 64-WA/LUB-8

The report discusses investigation of the performance of various dry-film lubricants and deep-grooved ball bearings operating under the temperature, speed, and load conditions expected in the pressure vessels of gas-cooled reactors. Bearings have rings and

balls of M-50 steel; and S-Monel cages properly treated with molydisulfide-type dry-film are found to be satisfactory for operation in dry helium for the conditions covered. Internal clearance and cage clearance are especially critical bearing-design criteria and need to prevent internal preloading. In addition, it is found that wear is relatively greater at low speeds and that the performance of dry film lubricated bearings is superior to that of unlubricated, cageless bearings.

APPENDIX III

RELATED INFORMATION

INTRODUCTION

Literature that is available on solid lubrication of gears and bearings is presented under the following three additional categories:

1. Materials
2. Specifications
3. Consolidated Literature

No effort was made to locate or designate powder lubricants. While the list below is not complete, it covers both development and commercial items and can be used as a guide in obtaining a variety of solid lubricants.

MATERIALS

Solid Composite Lubricants

Designation	Source
Armalon	DuPont de Nemours Company
Bar Temperature	Borden Corp.
Deva Metal	SKK Research Association
Duroid 5613	Rogers Corp.
Graphalloy	Graphite Metallizing
Ilkoloy AF-IL-M Series, W Series, C 10	Ilikon Corp.
Polymet A-G	The Polymer Corp.
RB Series, AP, CP, HP, HI, WGI	Westinghouse Electric Corp.
Raybearing R7600, R7000, R7800, R2959, Kynar Fluoray	Raybestos-Manhattan, Inc.
Rolute	Non-Metallic Bearings
Rulon Series	Dixon Graphite and Lubricants
Salox	Allegheny Plastics
Sintex	Booker Cooper Inc.
108-32, 114-14A, 101	The Boeing Company
Lubrite Series	Lubrite Div.; Meriman Bros. Inc.

Thin Film Lubricants

Designation	Source
Alpha-Molykote, E3C, X-106, M88	The Alpha Corp.
Calcium Fluoride-Barium Fluoride Mixtures; Lead Oxide; Sodium Silicate	NASA-Lewis
CLP 5940	CBS, Inc.
Dag 213, 223, 232	Acheson Industries, Inc.
Dynalube	National Process, Inc.
Emralon 310	Acheson Industries, Inc.
Electrofilm 4306	Electrofilm Corp.
Everlube Inlax 44, 620, 811 Series	Everlube Corp.
Fabroid	Micromatic Hone Corp.
Hi T-Lube	The Alpha Corp.
Magnalube D4821, D5261, D801	The Alpha Corp.
Lubeco 905 Series, N-350A	Lubeco, Inc.
MLF-5 Series	Midwest Research
Ore A Tape	Ore-Tube Company
Surf-Kote A1290	Hohman Plating
SP Polyimide	DuPont de Nemours Company
Tufftriding	Kolene Corp.
LA-2, LA5A	Linde Company
23A	Naval Aeronautical Materials Center

SPECIFICATIONS

Molybdenum Disulfide Powder Lubricant - MIL-M-7866(ASG), dated 22 December 1955.

Lubricant, Solid Film, Air Drying - MIL-L-23398A(ASG), dated 17 March 1964.

Plastic, Polyamide (Nylon) Rigid, Rods, Tubes and Flats, MIL-P-46060, dated 7 June 1963.

Lubricant, Solid, Film, Heat Cured, dated 12 February 1963.

CONSOLIDATED LITERATURE

Books

THE FRICTION AND LUBRICATION OF SOLIDS
Part I by Bowden and Tabor, Oxford University Press, 1954
Part II by Bowden and Tabor, Oxford University Press, 1964.

SOLID LUBRICANTS AND SURFACES
by Braithwaite, Pergamon Press, 1964.

ADVANCED BEARING TECHNOLOGY
by Bisson and Anderson, NASA SP-38, US Government Printing Office, 1964.

FRICTION AND WEAR
by Davies, Elsevier Publishing Company, 1959.

FRICTION AND WEAR OF MATERIALS
by E. Rabinowicz, 1965.

MECHANISMS OF SOLID FRICTION
by Bryant, Lavik, and Solomon; Elsevier Publishing Company, 1964.

LUBRICATION AND WEAR, PROCEEDINGS, INTERNATIONAL SYMPOSIUM
by Muster and Sternlicht; McCutchan Publishing Company, 1965.

HANDBOOK OF MECHANICAL WEAR
by Lipson and Colwell, University of Michigan Press, 1961.

SPACE MATERIALS HANDBOOK
by Gretzel, Rittenhouse, and Singletary; Addison-Wesley Publishing Company, 1965.

BIBLIOGRAPHY ON SOLID LUBRICANTS - NASA SP5037
by Tech. Utilization Division, NASA, 1966.

Bibliography

GEARS, BEARINGS AND LUBRICANTS FOR AEROSPACE APPLICATIONS: AN ANNOTATED BIBLIOGRAPHY
by Helen M. Abbott, Lockheed Missiles and Space Company, 1963.

PROCEEDINGS OF THE AIR FORCE-NAVY-INDUSTRY PROPULSION SYSTEMS LUBRICANTS CONFERENCE
by G. A. Beane and K. L. Berkey, Wright-Patterson AFB, Ohio, 1962.

Unclassified

Security Classification

DOCUMENT CONTROL DATA - R&D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Westinghouse Research Laboratories	2a. REPORT SECURITY CLASSIFICATION None 2b. GROUP NA

3. REPORT TITLE

Investigation of Solid Lubricants for Helicopter Transmissions

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Final Report - July 65 - November 1966

5. AUTHOR(S) (Last name, first name, initial)

Bowen, Paul H.

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
March 1967	80	9

8a. CONTRACT OR GRANT NO. Contract DA 44-177-AMC-307(T)	9a. ORIGINATOR'S REPORT NUMBER(S)
b. PROJECT NO. Task 1M125901A01410	USAAVLABS Technical Report 67-4
c.	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)
d.	66-9B3-LUBER-R4

10. AVAILABILITY/LIMITATION NOTICES

Distribution of this document is unlimited.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
None	U. S. Army Aviation Materiel Laboratories

13. ABSTRACT

The feasibility of using solid lubricants in bearings and gears ~~has been~~ was demonstrated as a satisfactory method for preventing catastrophic failure and for providing emergency operation of helicopter transmissions in the event of an oil lubrication failure.

Modified size-206 Conrad-type ball bearings with retainers of a glass reinforced polyimide, WRP-140, operated initially under a base line condition of oil lubrication for 40 hr. This was followed immediately by 0.5 hr of residual oil lubrication and 1.5 hr with no external lubrication at 14,000 rpm for various thrust loads of up to 800 lb without failure. Bearings with a silver alloy - Teflon composite, RB-HP-15 operated under similar oil and residual lubrication for 0.5 hr at loads of up to 450 lb before failure.

A conventional 12 diametral pitch (DP) AISI 9310 gear set, using a 6-in. gear and a 2-in. pinion with a 2.5-in. idler of WRP-140, operated under base line conditions of oil lubrication for 40 hr at approximately 14,000 rpm and at a tooth load of 160 lb [640 lb/in. tooth width (ppi)]. Similar gears with the same idler operated 0.5 hr with no external lubrication at a load of 1220 ppi (305 lb tooth load). With an RB-HP-15 idler, other gears were operated without lubrication for 0.5 hr at a load of 1220 ppi.

DD FORM 1 JAN 64 1473

Unclassified

Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
bearings composites friction gears helicopters lubricants oils powders solid lubricants thin films transmissions wear						

INSTRUCTIONS

1. **ORIGINATING ACTIVITY:** Enter the name and address of the contractor, subcontractor, grantee, Department of Defense activity or other organization *(corporate author)* issuing the report.

2*a*. **REPORT SECURITY CLASSIFICATION:** Enter the overall security classification of the report. Indicate whether "Restricted Data" is included. Marking is to be in accordance with appropriate security regulations.

2*b*. **GROUP:** Automatic downgrading is specified in DoD Directive 5200.10 and Armed Forces Industrial Manual. Enter the group number. Also, when applicable, show that optional markings have been used for Group 3 and Group 4 as authorized.

3. **REPORT TITLE:** Enter the complete report title in all capital letters. Titles in all cases should be unclassified. If a meaningful title cannot be selected without classification, show title classification in all capitals in parenthesis immediately following the title.

4. **DESCRIPTIVE NOTES:** If appropriate, enter the type of report, e.g., interim, progress, summary, annual, or final. Give the inclusive dates when a specific reporting period is covered.

5. **AUTHOR(S):** Enter the name(s) of author(s) as shown on or in the report. Enter last name, first name, middle initial. If military, show rank and branch of service. The name of the principal author is an absolute minimum requirement.

6. **REPORT DATE:** Enter the date of the report as day, month, year; or month, year. If more than one date appears on the report, use date of publication.

7*a*. **TOTAL NUMBER OF PAGES:** The total page count should follow normal pagination procedures, i.e., enter the number of pages containing information.

7*b*. **NUMBER OF REFERENCES:** Enter the total number of references cited in the report.

8*a*. **CONTRACT OR GRANT NUMBER:** If appropriate, enter the applicable number of the contract or grant under which the report was written.

8*b*, 8*c*, & 8*d*. **PROJECT NUMBER:** Enter the appropriate military department identification, such as project number, subproject number, system numbers, task number, etc.

9*a*. **ORIGINATOR'S REPORT NUMBER(S):** Enter the official report number by which the document will be identified and controlled by the originating activity. This number must be unique to this report.

9*b*. **OTHER REPORT NUMBER(S):** If the report has been assigned any other report numbers *(either by the originator or by the sponsor)*, also enter this number(s).

10. **AVAILABILITY/LIMITATION NOTICES:** Enter any limitations on further dissemination of the report, other than those imposed by security classification, using standard statements such as:

(1) "Qualified requesters may obtain copies of this report from DDC."

(2) "Foreign announcement and dissemination of this report by DDC is not authorized."

(3) "U. S. Government agencies may obtain copies of this report directly from DDC. Other qualified DDC users shall request through ________________."

(4) "U. S. military agencies may obtain copies of this report directly from DDC. Other qualified users shall request through ________________."

(5) "All distribution of this report is controlled. Qualified DDC users shall request through ________________."

If the report has been furnished to the Office of Technical Services, Department of Commerce, for sale to the public, indicate this fact and enter the price, if known.

11. **SUPPLEMENTARY NOTES:** Use for additional explanatory notes.

12. **SPONSORING MILITARY ACTIVITY:** Enter the name of the departmental project office or laboratory sponsoring *(paying for)* the research and development. Include address.

13. ABSTRACT: Enter an abstract giving a brief and factual summary of the document indicative of the report, even though it may also appear elsewhere in the body of the technical report. If additional space is required, a continuation sheet shall be attached.

It is highly desirable that the abstract of classified reports be unclassified. Each paragraph of the abstract shall end with an indication of the military security classification of the information in the paragraph, represented as *(TS)*, *(S)*, *(C)*, or *(U)*.

There is no limitation on the length of the abstract. However, the suggested length is from 150 to 225 words.

14. KEY WORDS: Key words are technically meaningful terms or short phrases that characterize a report and may be used as index entries for cataloging the report. Key words must be selected so that no security classification is required. Identifiers, such as equipment model designation, trade name, military project code name, geographic location, may be used as key words but will be followed by an indication of technical context. The assignment of links, rules, and weights is optional.